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DESIGNING HEATING AND VENTILATING SYSTEMS

CHARLES A. FULLER

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Designing Heating and Ventilating Systems

The practical application of the engineering rules and
formulas in every day use, in laying out steam,
hot water, furnace and ventilating equipment
for buildings of all kinds, presented in
a simple and easily understandable
manner. Adopted from lecture
courses given by the author
before Y. M. C. A. and
other classes

By CHARLES A. FULLER
Consulting Engineer

UNIV. OF
CALIFORNIA

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PREFACE

This book has been developed from a series of evening lecture courses given at several institutions, before audiences of varying experience and education in the heating line. It enters into the detailed use of exactly the same methods that the most proficient engineer uses in determining the sizes and proportions of equipment for everyday work and the problems that confront him for the first time. This basic engineering has been made so simple that salesmen and practical steamfitters have gained a sufficient grasp at the lectures without the careful study the reader must give to understand how to approach the solution of the problems that are presented by the many kinds of work involved in heating and ventilation.

In all explanations the heat unit, foot pound and such measures of the engineer are so used that those who attended the lectures and gave their attention showed a marked proficiency in their use. The reader who will apply the information to work of which he knows something is expected to be equally benefitted and more experience and more frequent reference to the work will be sure to increase his proficiency. Its origin was inspired by the necessity of an up-to-date text book which would involve both the theoretical, as well as the practical side of the subject and one which would be really understood by the less techincal mind.

The theory of both heating and ventilating is handled in such a manner as to eliminate as far as possible, the most difficult mathematical computations and expressions, and at the same time, is handled in a thorough and exhaustive manner.

The book is well adapted for colleges, technical schools, vocational and trade schools and as a text book for lecture courses on this subject. It is also very suitable as a reference book for consulting engineers, architects and contractors, as it is developed along the lines of the most recent practice in all

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classes of buildings from actual experience of the author in one of the largest consulting engineering offices in New York.

The author expresses his appreciation of the kindness of those manufacturers who furnished illustrations of their apparatus to make the work more complete.

C. A. FULLER

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DESIGNING HEATING AND VENTILATING SYSTEMS

CHAPTER I

THE HEAT UNIT

THE thermometer is an instrument for measuring the intensity of heat. There are several different standards adopted in various countries, the two most common of which are the Fahrenheit, or the F. scale, and the Centigrade, or the C. scale. The unit generally adopted in this country for the measurement of the quantity of heat is known as the British thermal unit and designated by the letters B. t. u.

One B. t. u. is the amount of heat necessary to raise the temperature of 1 lb. of water 1 deg. on the Fahrenheit scale. The temperature at which this is given is from 39 deg. to 40 deg. Some authorities differ slightly on this point and give the temperature at which this should be taken as 63 deg. F. to 64 deg. F. The difference in the amount of heat at these two points, however, is so slight that for practical purposes the temperature need not be considered and the adopted standard for the heat unit may be given as simply the quantity of heat necessary to raise 1 lb. of water 1 deg. F.

The reason for selecting the temperature of 39 deg. is that at this point water reaches its maximum density. As the temperature increases above this point the volume also increases proportionately. At 39 deg. F. the weight of water per cubic foot is 62.424 lb. At 212 deg. F. its weight per cubic foot is 59.76 lb.

The heat unit which corresponds to the B. t. u. on the Centigrade scale or the French unit is designated as one Calorie.

The Calorie is the amount of heat necessary to raise the temperature of one Kilogram of water 1 deg. C. As one kilogram of water is equal to 2.2 lb. and 1 deg. C. corresponds to 1.8 deg. on the Fahrenheit scale, one Calorie equals 2.2 x 1.8 or 3.96 B. t. u.

Relation Between Thermometers

It is sometimes necessary in making certain calculations to transfer readings from one scale to another. In order to determine this it will be necessary to determine the mathematical relations between the various scales. To illustrate this two thermometers are shown in Fig. 1 designated as F. and C. and representing respectively the Fahrenheit and the Centigrade scale. The two *fixed points*, namely the freezing point and the boiling point of water, are 32 deg. and 212 deg. on the former and 0 deg. and 100 deg. on the latter. On the F. scale there are 212-32, or 180 divisions, between the fixed points, and correspondingly 100 divisions between the fixed points on the C. scale. Therefore, the Fahrenheit divisions are $\frac{100}{180}$ or $\frac{5}{9}$ of the divisions or degrees on the Centigrade thermometer.

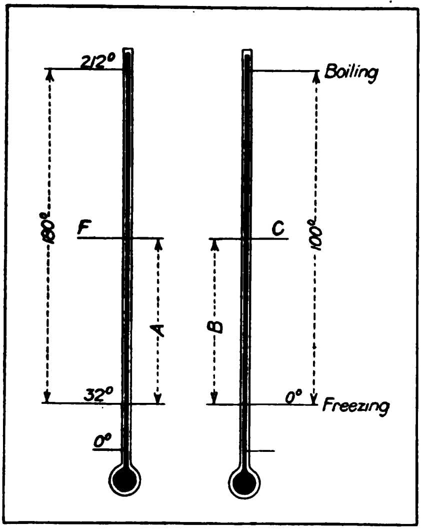


Fig. 1—Diagram of Fahrenheit and Centigrade Scales

Assume some arbitrary temperature represented by the line F C through the two thermometers. Let the distance or the number of degrees of this assumed temperature above the freezing on the F scale be represented by A and the corresponding distance on the C scale be represented by B. The divisions on F. being $\frac{5}{9}$ of the divisions on C, we can say that $\frac{5}{9}$ of A equals B, or $A = \frac{9}{5}B$. But in order to get the actual reading of this assumed temperature on the F scale above zero, which we will call F, it is necessary to add 32 deg. to the distance A, or we can say $F = 32 + A$. Transposing the 32 to the opposite side of the equation $A = F—32$. It has already been shown however that $A = \frac{5}{9}B$, which we will now designate as C on the reading on the Centigrade scale. Therefore we can establish the formula $\frac{9}{5} C = F—32$. Transposing the $\frac{9}{5}$ to the opposite side of the equation we have

$$C = \frac{5}{9}(F—32). \qquad (1)$$

By transposition, this formula may also be expressed thus:

$$F = \frac{9}{5}C + 32 \qquad (2)$$

By use of these two formulas readings may be readily changed from one scale to another.

Assume a reading of 113 deg. on the Fahrenheit scale and it is desired to find the corresponding reading on the Centigrade scale. As F is known and C is unknown we will use the formula which gives the value of C in terms of F or formula No. 1.

$$C = \frac{5}{9} (F—32). \quad F = 113.$$

Therefore

$$C = \frac{5}{9}(113—32) = \frac{5}{9} \times 81 = 45 \text{ deg.}$$

Problems

(1) Transpose a reading of 80 deg. C. to the Fahrenheit scale.

(2) At what temperature will the two thermometers have the same reading?

Solution to Problems

PROBLEM No. 1

Substituting the known value of C, which in this case is 80, in formula (2), we have F = 9/5 of 80 + 32 = 176. Therefore 80 deg. on Centigrade scale is equal to 176 deg. on the Fahrenheit scale.

PROBLEM 2

Take formula 1, viz. C = 5/9 (F—32). Multiply each side by 9/5 we have 9/5 C = F—32. But F = C. Therefore 9/5 C = C—32. Or subtracting C from each side we have 4/5 C = —32. Clearing each side of fractions gives C = —40. Therefore —40 deg. F. is the same temperature as —40 deg. C.

Absolute Zero

If the temperature of any gas is increased and the pressure of the gas kept constant the volume of the gas will increase. If the gas is what is known as a perfect gas its rate of increase in volume will exactly equal its rate of increase in temperature. All gases that are dealt with in mechanical engineering such as steam, air, oxygen, hydrogen, etc., are very nearly perfect gases and follow this law closely. It has been found by experiment that a perfect gas will expand, for each degree Centigrade of increase in temperature 1/273 part of its volume at 0 deg. C. The reverse of this is also true; that for each degree the gas is cooled it will contract 1/273 part of its volume at 0 deg. Centigrade.

If this process of cooling and contracting were carried on to a point 273 deg. below 0 deg. C., theoretically the volume of the gas would have disappeared entirely and the gas would be absolutely void of heat. This theoretical point on the Centigrade scale is called Absolute Zero.

To determine this point on the Fahrenheit scale refer to formula 2.

$$F=\frac{9}{5}C+32. \qquad C=\frac{5}{9}273 \text{ deg.}$$

$$F=\frac{9}{5}(-273)+32=-492+32=-460$$ deg. or 460 deg. below zero.

This scale is usually designated by T and all formulas involving only one temperature should be calculated on this basis. If the formula involves the difference between two temperatures it is not necessary to use the Absolute scale, as the result would be the same whether the Absolute or the regular scale were used.

CHAPTER II

THE HEATING VALUE OF COAL

THE universal source of heat for power and heating purposes is derived from coal. In some western sections of the country where coal is very expensive, crude oil is used quite extensively, but in the eastern and central states, it is considerably more expensive than coal when compared on a B. t. u. basis. The number of B. t. u. per pound of coal varies with the quality of coal and the location from which it is mined. The lowest grades of anthracite coal give about 11,000 B. t. u. per pound and the heating value ranges from this up to 14,500 for the highest grades. Bituminous coal, which is more commonly known as soft coal, gives from 9,000 to 13,000 B. t. u. per pound.

For estimating purposes when the actual heating value of the coal to be used is not known an average value of 13,000 B. t. u. may safely be assumed for anthracite and 11,000 B. t. u. for bituminous coal. The heating value of dry wood is usually taken at about 7,500 B. t. u. per pound.

To estimate the size of storage space the weight of coal may be taken at about 54 lb. per cu. ft. for hard coal and 50 lb. per cu. ft. for soft coal.

The method of determining the heating value of coal is by means of a calorimeter, one standard form of which is shown in Fig. 2. A sample of coal to be tested of a given weight is first thoroughly dried. After the drying process it is again weighed to determine the amount of moisture present. This amount of moisture is always expressed as a percentage of the total weight.

If A represents the weight of the coal before the moisture is removed and B represents the weight of the same volume of coal after the moisture is removed, then A —B equals the actual weight of the moisture $(A - B) \div B =$ percentage of moisture.

In Fig. 2 A is a closed metal cylinder in which a quantity of the coal to be tested and of which the weight has been accurately determined is placed. The space B is filled with water which is also accurately weighed. The receptacle C, which contains the water, is thoroughly insulated to prevent the loss of heat from the water to the surrounding air. The thermometer T is immersed in the water through an opening in the cover.

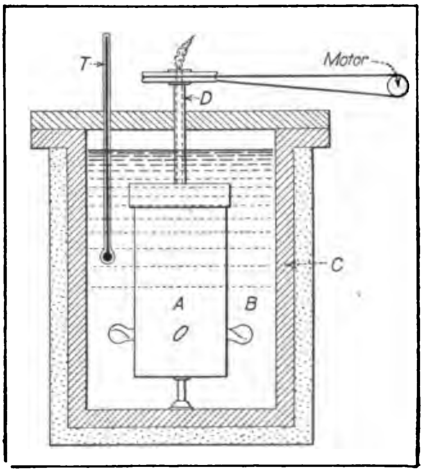


FIG. 2—SECTION OF CALORIMETER

The cylinder A is slowly rotated by means of a small motor during the test and the paddle wheels on the side of the cylinder keep the water in motion so as to produce a uniform temperature. The coal is pulverized and mixed with a chemical, usually an oxide of sodium, and the mixture is ignited by means of a piece of hot metal dropped into the cylinder through the hollow spindle at the top or by means of an electric spark. The burning of the coal raises the temperature of the water which is recorded by the thermometer. The thermometer should be observed until the temperature begins to fall and this maximum temperature recorded.

Let A = weight of coal.
B = weight of water.
T_1 = initial temperature.
T_2 = final temperature.

Then $B \times (T_2 - T_1)$ = B. t. u. given up to the water.
$B (T_2 - T_1) \div A$ = B. t. u. per pound of coal.

The amount of heat taken up by the metal in the apparatus must also be taken into account in the computation. This is usually given as the constant of the calorimeter.

The approximate rise in temperature is usually first determined and the temperature of the water adjusted at the start of the test so that it is about ½ the amount of this rise below the temperature of the room. This will practically eliminate any error due to loss of heat to the surrounding air, as a certain quantity of heat will be taken up by the water from the outside air during the first half of the test and this same quantity of heat will be given out again to the air from the water during the last half and thus the loss will be neutralized.

CHAPTER III

RELATION OF HEAT AND WORK

HEAT is a form of molecular energy and by proper application it may be changed into the form of work. To illustrate, a certain quantity of coal is burned under a boiler and heat is produced by the combustion. A portion of this heat is transmitted through the boiler shell to the water in the boiler. This heat transforms the water into steam under pressure. The steam now possesses the same energy that was formerly present in the unburned coal. The steam is transmitted to the cylinder of an engine and by expanding and giving up its heat produces motion, thereby doing a definite quantity of work.

This may be expended by raising a weight through a certain distance, such as raising an elevator to the top of a building. This elevator now in its final position possesses the same energy that was first given up by the burning of the coal only in a different form. This energy is defined as potential energy or energy of position. The elevator could in turn transmit energy still farther by falling to its original position. The energy would be dissipated by heating the body which the elevator would strike in its fall.

If there had been no loss to other sources in these various transformations, the amount of heat given up by the elevator due to impact would be equal to the heat originally possessed by the coal. It can therefore readily be seen that heat and work are mutually interchangeable. This being true there must be some relation between unit quantities.

The unit of work is 1 ft. lb., which means the amount of work or energy necessary to raise 1 lb. of any matter through a height of 1 ft. The relation between heat and work is denoted as the mechanical equivalent of heat. It has been determined by a series of experiments that 1 B. t. u. is equivalent to 778 ft. lb. of work. If 1 B. t. u. could be converted into work without any loss it would be capable of raising 1 lb. of mass through a height of 778 ft. or 778 lb. through a height of 1 ft.

Horse Power

Power is the rate at which work is done and involves the time element. It would require no more work to raise the 1-lb. weight through a distance of 778 ft. in 1 min. of time than in 1 hr. of time but it would require 60 times as much power. Therefore, the quotient obtained by dividing foot pounds of work by the time it takes to accomplish this work expresses the rate or power necessary.

The unit adopted for this in engineering practice is one horse power, which is an arbitrary quantity of 550 ft. lb. per second or 33,000 ft. lb. per minute. An engine which will develop one horse power is capable of lifting 33,000 lb. 1 ft. per minute.

The term "horse power" when applied to a boiler has no definite relation to the term "horse power" as applied to an engine or any prime mover. As a boiler does no actual work, the term "boiler horse power" can be designated only as a certain number of heat units per unit of time. The standard adopted for a boiler horse power is 34.5 lb. of water at a temperature of 212 deg. F. converted into steam at 212 deg. per hour. This expressed in B. t. u. is the amount of heat necessary to evaporate 1 lb. of water at 212 deg. into steam at 212 deg. and is represented by the latent heat of steam which will be explained later under the heading of steam tables, and is equal to approximately 966 B. t. u. Therefore, one boiler horse power is equal to $34.5 \times 966 = 33{,}327$ B. t. u. per hour. One engine horse power may also be expressed in B. t. u. per hour as follows:

One h.p. = 33,000 ft. lb. per minute or $33{,}000 \times 60 = 1{,}980{,}000$ ft. lb. per hour.

One B.t.u. is equivalent to 778 ft. lb.

$$\text{Therefore, 1 h.p.} = \frac{1{,}980{,}000}{778} = 2545 \text{ B.t.u. per hour.}$$

It will be seen that this is only about 8 per cent. of the B.t.u. represented by one boiler horse power. The reason for this apparently large difference in ratings is due to the fact that the average engine utilizes only about 8 per cent of the actual heat in the steam which it consumes. The percentage of heat which it utilizes is dependent upon the type and efficiency of the engine. The largest proportion of the heat is lost in the exhaust steam which still retains the latent heat. For this reason the exhaust steam from an engine is extremely valuable for heating purposes and in up-to-date plants is always utilized for this purpose when heating is necessary.

CHAPTER IV

LOSS OF HEAT FROM BUILDINGS

HEAT is lost from a building in three ways, first by conduction, second by convection and third by radiation.

To illustrate the first method, place one end of an iron bar in a flame and observe the rise in temperature of the bar at various distances from the flame. This heat is carried from one point of the bar to another by conduction. The speed with which the heat will be conducted from one point to another in any body varies with the kind of material of which it is composed and is also directly proportional to the difference between the temperatures of the two points. The direction of flow of heat is always from the higher to the lower temperature.

For example, the inside of the wall of a building being in direct contact with the warm air of the room is at a considerably higher temperature than the outside which is in direct contact with the colder outside air. Heat therefore passes through the wall by conduction to the outside and is carried away by the passage of air over the outside surface.

Heat Convection

The loss of heat by convection is the loss of heat through contact with air or the heat carried away by the air leaving the room through the walls, around windows, doors, etc. The air that leaves the room must be replaced from some source and this is usually the cold outside air. The number of B. t. u. lost per hour through this source is represented by the number of B. t. u. necessary to raise the temperature of 1 cu. ft. of air from the temperature of the incoming air to the temperature of the outgoing air multiplied by the number of cubic feet of air leaving the room per hour.

Radiation

Heat lost or given off by radiation is more properly termed radiant heat. Radiant heat may be noticed very perceptibly when one is standing in front of a hot flame of fire by its effect on the face or any exposed portion of the body. Place any opaque surface between the face and the flame and the burning sensation ceases. Place a piece of glass or any transparent surface between the face and the flame and the burning sensation will still be felt as with the glass removed.

This indicates that radiant heat is transmitted in the same manner as light and objects which are transparent to the rays of the light are also transparent to radiant heat. This transmitting of radiant heat through a body or transparency of heat rays is called in engineering work diathermancy.

The total heat lost from a room is usually a combination of all the three foregoing methods and these must be considered in determining the heat loss from a building. It is not necessary, however, to determine each separately. The loss by conduction and radiation are usually combined under one heading and expressed in B. t. u. per square foot of wall surface for each degree difference in temperature between the two sides of the wall. These constants are called factors of exposure.

Authorities differ somewhat as to the values of these factors and the accompanying table is compiled by averaging the values given by several good authorities.

TABLE I.

Factors of Heat Loss for Walls

	B. t. u. per square foot per degree difference per hour		
Thickness of Wall, In.	Brick	4-in. Additional Stone Face	Concrete or Sand Stone
4	0.60	....	0.65
8	0.42	0.38	0.48
12	0.30	0.26	0.41
16	0.24	0.22	0.35
20	0.21	0.20	0.32
24	0.19	0.18	0.29
28	0.17	0.16	0.26
32	0.15	0.14	0.24
36	0.13	0.12	0.22
40	0.12	0.11	0.20

The constants referred to are for walls plastered on the inside. For walls not plastered these constants should be increased 10 per cent. If the walls are furred and plastered, thus providing an air space, the constants may be decreased 5 per cent.

Other wall surfaces may be taken as follows:

Frame buildings, lathed and plastered on inside, with outside of wall as follows:

TABLE II

Ordinary clapboards	0.48
Same with paper lining	0.34
Same with ¾-in. sheathing	0.30
Same with ¾-in. sheathing and paper lining	0.25
Unlined corrugated iron	0.80
Same on f and g boards	0.40
Glass Surfaces	
Single windows	1.0
Double windows	0.5
Single skylights	1.0
Double skylights	0.5
Doors considered same as windows.	
Roof Surfaces	
Concrete, with cinder fill	0.50
Slate, no sheathing	0.74
Slate, with sheathing	0.38
Patent tar and gravel roof	0.28
Partitions	
Lath and plaster on studding	0.60
Lath and plaster both sides on studding	0.36
Floors	
Single wood flooring, no plaster beneath	0.28
Single wood flooring, plaster beneath	0.20
Double wood flooring, no plaster beneath	0.24
Double wood flooring, plaster beneath	0.18
Concrete on ground	0.40
Wood near ground	0.20
Dirt (no floor)	0.30
Temperature of ground taken at 50°.	
Ceilings	
Lath and plaster, no floor above	0.40
Lath and plaster, wood floor above	0.36

Air Change

The amount of air passing out of a room through leakage is more or less theoretical and is difficult to determine with any degree of accuracy. It is dependent on the type of construction of the building, the tightness of windows, sashes, etc. It is also dependent on the size of the room or the ratio of the volume of the room to the square foot of exposed wall and glass surface. Open fireplaces, stairs, outside doors, etc., also tend to increase the number of air changes per hour and should be taken into consideration. It is generally assumed that average-sized rooms up to 10,000 cu. ft. contents under ordinary conditions will have one complete air change every hour. The following figures may be used with safety but should be modified to suit local conditions.

Entrance halls, vestibules, etc., three changes per hour.

Living rooms, with open fireplaces, two changes per hour.

Bedrooms, small office rooms, etc., one change per hour.

Large offices, lofts, etc., one-half to one change per hour depending on the amount of exposure.

Another method of estimating the air change per hour is by first determining the average number of persons to occupy the room. Allow about 25 cu. ft. of air per person per minute. This will give the total number of cubic feet of air that should be supplied to give good ventilation and can be obtained up to certain limits by partly opening windows and at the same time required room temperature will be maintained. This quantity divided by the volume of the room gives the number of air changes per hour.

B. T. U. Required for Air Change

To determine the number of B. t. u. necessary for the assumed air change the specific heat of air must be taken into account which is the number of B. t. u. necessary to raise 1 lb. of air 1 deg. There are two values given for this, namely, specific heat at constant volume and specific heat at constant pressure. As has been previously stated, when any gas is heated its volume is increased if the pressure is maintained constant. Correspondingly, if the volume is kept constant, and the gas is heated the pressure increases. To raise the temperature of 1 lb. of air 1 deg. under constant pressure requires more heat than under constant volume as in the former case, additional work or energy must be expended in increasing the volume of the air as well as raising the temperature.

The specific heat air is as follows:

At constant volume, 0.1685.

At constant pressure, 0.2375.

When the air in a room is heated the pressure remains constant and the volume increases, the excess air passing out through leakage openings. For this reason the specific heat at constant pressure is used. One pound of air at 32 deg. F. and atmospheric pressure occupies 12.14 cu. ft. of space and 1 cu. ft. of air under the same conditions weighs 0.0807 lb. To raise 1 cu. ft. of air 1 deg. requires $0.0807 \times 0.2375 = 0.0192$ B. t. u. or approximately 0.02 B. t. u. One B. t. u. will raise $1 \div 0.0192$ cu. ft. = 52 cu. ft. of air 1 deg. at 32 deg. F. At 70 deg. this figure is 56 cu. ft. and is usually taken at 55 cu. ft. for average conditions.

To combine the foregoing factors for wall, glass and air changes, let

W = Net exposed wall surface in square feet.
K_w = Factor of exposure for wall.
G = Total square feet exposed glass surface.
K_g = Factor of exposure for glass.
V = Volume of room in cubic feet.
N = Number of air changes, assumed.
T = Difference in temperature between lowest outside weather conditions and desired room temperature (usually 70 deg.)

Then $W \times K_w$ = B. t. u. loss through walls for each degree difference in temperature.

$G \times K_g$ = B. t. u. loss through glass for each degree difference in temperature.

$V \times N \times 0.02$ = B. t. u. loss through air change for each degree difference.

Adding $(W \times K_w) + (G \times K_g) + (V \times N \times 0.02)$ = total B. t. u. loss from room for each degree difference.

$(W \times K_w) + (G \times K_g) + (V \times N \times 0.02) \times T$ = total B. t. u. loss from room per hour. (1)

This represents the number of B. t. u. that must be supplied to the room per hour in order to maintain the desired room temperature.

This formula gives the loss for ordinary exposure and should be increased for extreme conditions as follows:

Rooms with north exposure, add 10 per cent.
Rooms with west exposure add 10 per cent.
Buildings heated during the daytime only, add 15 per cent.
Buildings heated at long intervals only, add 25 per cent.
Rooms with ceiling heights over 12 ft., add 2 per cent. for each foot in height above 12 ft.

Unheated spaces such as attics, entrance halls, cellars, etc., are usually estimated at 30 deg. The difference between the room temperature of 70 deg. and 30 deg. or 40 deg., times the factor for the separating wall, times the number of square feet of this wall will give the B. t. u. lost through this wall or partition and must be included in the total loss.

Problem

Given a room of dimensions as shown in Fig. 3 find the number of B. t. u. lost per hour when the outside temperature is zero and the room temperature 70 deg. F. This room has a 12-in. brick wall plastered on the inside. Room A is an unheated hall and the wall between is lathed and plastered on both sides. All the other adjoining rooms are heated. The room faces north.

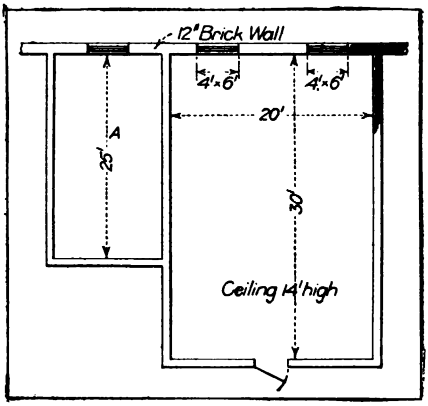


Fig. 3—PLAN OF ROOM IN PROBLEM

Total exposed wall $=20\times14=280$ sq. ft.

Total exposed glass $=4\times6=24\times2=48$ sq. ft.

Net exposed wall $=280—48=232$ sq. ft.

Wall factor for 12-in. brick wall is 0.30.

Glass factor is 1.

Contents or volume of room $=20\times30\times14=8400$ cu. ft.

Assume one air change per hour.

T, the difference in temperature between the inside and outside $=70$ deg. F.

Applying the foregoing values in formula:

$$\begin{aligned}\text{B. t. u.} &= [(W \times Kw) + (GKg) \\ &\quad + (V \times N \times 0.02)]T \\ &= [(232 \times 0.30) + (48 \times 1) \\ &\quad + (8400 \times 1 \times 0.02)] \times 70 \\ &= (69.6 + 48 + 168)\ 70 \\ &= 285.6 \times 70 = 19{,}992.\end{aligned}$$

This is the number of B. t. u. lost per hour through the outside wall.

If the temperature of the unheated hallway is assumed as 30 deg., 70—30 = 40 deg. difference.

25 × 14 = 350 sq. ft. wall.

Factor for lath and plastered partition is 0.36.

350 × 0.36 × 40 = 5040 B. t. u. lost through side wall to hall.

19,992 + 5040 = 25,032 B. t. u.

10 per cent. of the foregoing for north exposure = 2503.

14-ft. ceilings being 2 ft. over 12 ft. and 2 per cent. for each foot over 12 is 4 per cent.

4 per cent. of 25,032 = 1001.

25,032 + 2503 + 1001 = 28,536, total B. t. u.

CHAPTER V

CLASSIFICATION OF HEATING SYSTEMS

Properties of Saturated Steam

THE preceding chapters have been devoted entirely to the discussion of the heat unit and the question of determining the heat losses from buildings under various conditions. The next question in order is that of supplying heat to counteract these losses. The various methods of accomplishing this with steam and hotwater heating systems will now be taken up. Each of these may be subdivided into several classes.

No fixed rules can be given to follow closely in the determination of which of these two systems to adopt for any particular building. This can only be determined by a careful study of local conditions. In large buildings, factories, etc., the question should be considered from the standpoint of economy of operation as well as the various advantages and disadvantages that may accompany either of the systems.

For residence work of moderate size, the hot-water system is generally considered to be the better of the two, although there are certain conditions where this would not be true, and all questions bearing on the subject should be carefully considered before any final selection is made.

The advantages and disadvantages of the two systems may be summed up as follows:

A more uniform room temperature is maintained due to the large quantity of water in the system and fluctuations of fire conditions do not produce a correspondingly rapid fluctuation in the room temperature.

Healthier air conditions are produced with the hot-water system due to the lower temperature of the radiation and a higher relative humidity.

The temperature of the water may be varied through a wide range, thus providing a means of controlling the room temperature for varying outside weather conditions. In large plants, this may be placed entirely under the control of the engineer. This eliminates the tendency on the part of tenants and occupants of the rooms to raise windows in overheated rooms which occurs with steam systems and thus correspondingly increases the coal consumption. The hot-water system requires slightly less coal consumption per season than steam.

Larger radiators are necessary due to the lower temperature of the water.

The circulation is slow, requiring a comparatively long time to raise the room temperature. Conversely, if a radiator is shut off it will require a considerable length of time for the water in the radiator to cool and allow the room temperature to drop.

There is considerable danger from freezing in exposed locations.

The system is more expensive to install than a steam system on account of the increased amount of radiation necessary.

Both steam and hot-water systems may be subdivided as follows:

Direct, in which the radiator is placed within the room to be heated.

Indirect, in which the radiator is placed in the basement with a cold-air duct leading from the outside to a sheet-metal casing surrounding the radiator and a warm-air duct leading from this casing to the room to be heated.

Direct-indirect, which is a combination of the other two systems. The radiator is placed within the room to be heated and is provided with a cold-air passage leading from the outside to the under side of the radiator.

Hot blast, in which the heating stacks or coils are placed in the basement or in a separate room and cold air is forced by means of a blower through these coils, heated to the required temperature and distributed to the various rooms by means of ducts.

For the last three systems mentioned more radiation must be allowed for maintaining the required room temperature and it must be treated differently. The method of estimating the amount of radiation for these systems will be given later under the respective headings.

Properties of Saturated Steam

In order to understand properly the proportioning of radiators, supply mains, etc., for steam systems, the nature and properties of steam under various conditions should first be thoroughly investigated. For this purpose a table giving the properties of saturated steam under different pressures is given herewith.

Column 1 gives the pressure of steam in pounds per square inch above atmosphere, commonly called gauge pressure, and vacuum in inches of mercury below atmosphere.

Column 2 gives the temperature at which water boils when under the corresponding pressure.

Column 3 gives the heat in the liquid, which is the number of B. t. u. necessary to raise 1 lb. of water from freezing temperature, 32 deg. F., to the corresponding temperature in the table.

Column 4 gives the latent heat, which is the number of B. t. u. necessary to change 1 lb. of water at boiling temperature into steam at atmospheric pressure.

Column 5 gives the total heat or number of B. t. u. per lb. of steam at the corresponding temperature in column 2.

Column 6 gives the volume of 1 lb. of steam at the corresponding pressure.

Problem

How many B. t. u. will be required to transform 15 lb. of water at a temperature of 100 deg. F. into steam at 10 lb. pressure per square inch.

Solution

The total heat of steam at 10 lb. pressure is shown in column 5 of the table to be 1154 B. t. u. As the water to be transformed to steam is at a temperature of 100 deg. F. it will contain 100—32 = 68 B. t. u. per lb. to be subtracted from the total heat. Therefore 1154—68 = 1086, which is the number of B. t. u. required to change 1 lb. of water at 100 deg. F. into steam at 10 lb. pressure and 1086 × 15 = 16,290, the number of B. t. u. required for the transformation of 15 lb. of water. The total heat transferred from steam at various pressures to the atmosphere at various temperatures may also be ascertained from the table, and the condensation of steam per square foot of radiating surface per hour, when the B. t. u. transmission per square foot of surface is known.

TABLE III

Properties of Saturated Steam

I Press. lb. per sq. in. or vac'm in mercury.	II Temperature	III Heat of liquid. B. t. u.	IV Latent heat, B. t. u.	V Total heat, B. t. u.	VI Volume of 1 lb. steam, cu. ft.
— 12	137.0	105.0	1019.0	1124.0	135.00
— 10	160.0	128.0	1003.0	1131.0	78.30
— 8	175.0	143.0	992.0	1135.0	55.90
— 6	187.0	155.0	984.0	1139.0	43.60
— 4	197.0	165.0	977.0	1142.0	35.80
— 2	205.0	173.0	971.0	1144.0	30.60
0	212.0	180.9	965.7	1146.6	26.36
1	215.0	184.0	964.0	1148.0	25.00
2	219.0	188.0	961.0	1149.0	23.00
3	222.0	191.0	959.0	1150.0	22.30
4	224.0	193.0	957.0	1150.5	21.20
5	227.0	196.0	955.0	1151.0	20.16
10	239.0	208.0	946.0	1154.0	16.30
15	249.0	218.8	939.3	1158.1	13.70
20	258.7	228.0	932.5	1161.0	11.85
25	266.7	236.2	927.1	1163.3	10.36
30	273.9	243.5	922.0	1165.5	9.34
35	280.5	250.2	917.3	1167.5	8.45
40	286.5	256.3	913.0	1169.3	7.73
45	292.2	262.1	909.0	1171.1	7.11
50	297.5	267.5	905.2	1172.7	6.61
55	302.4	272.6	901.6	1174.2	6.16
60	307.1	277.2	898.4	1175.6	5.77
65	311.5	281.8	895.1	1176.9	5.43
70	315.8	286.1	892.1	1178.2	5.13
75	319.8	290.3	889.1	1179.4	4.86
80	323.7	294.3	886.3	1180.6	4.63
85	327.4	298.1	883.6	1181.7	4.41
90	330.9	301.8	881.0	1182.8	4.20
95	334.4	305.4	878.5	1183.9	4.02
100	337.6	308.9	876.0	1184.9	3.83
110	343.9	315.4	871.4	1186.8	3.57
120	349.8	321.5	867.1	1188.6	3.33
130	355.0	327.5	863.0	1190.3	3.10
140	360.0	333.5	859.1	1191.9	2.92
150	365.7	338.3	855.4	1193.4	2.75

CHAPTER VI

HEAT TRANSMISSION THROUGH RADIATORS AND COILS

THE standard types of radiators are made in one, two, three and four column sections, as shown in Figs. 4 to 7. These types are made in heights of 22 in., 26 in., 32 in. and 38 in. to suit various conditions. For instance, they are made with high or low legs and also without legs where they are required to be hung on brackets. In addition to these, there is also the window radiator shown in Fig. 8 and made in heights of from 12 to 18 in. and the wall radiator shown in Fig. 9.

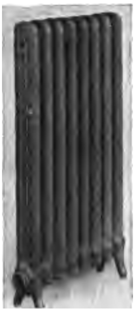


FIG. 4—SINGLE COLUMN

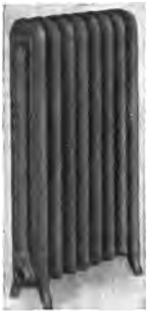


FIG. 5—TWO COLUMNS

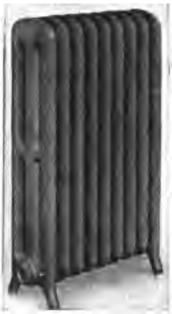


FIG. 6—THREE COLUMNS

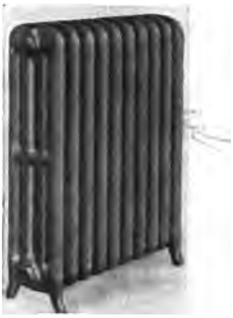


FIG. 7—FOUR COLUMNS

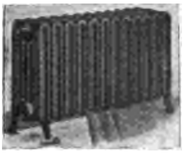


FIG. 8—WINDOW RADIATOR

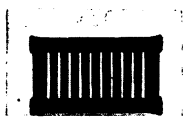


FIG. 9—SECTION OF WALL RADIATOR

The number of square feet of radiating surface per section for these various types can be obtained from any of the manufacturers' catalogues, such as the American Radiator Co., the United States Radiator Corp., or the H. B. Smith Co. The following table, however, giving the number of square feet per section, conforms very closely to the ratings given by the above companies:

TABLE IV

Type of Radiator	Square feet per section 22 in.	26 in.	32 in.	38 in
1 col.	1¾	2	2½	3
2 col.	2⅓	2⅔	3⅓	4
3 col.	3	3¾	4½	5
4 col.	4	5	6½	8

Wall radiators are made in several sizes ranging from 5 to 9 sq. ft. per section. The construction is such that any number of sections can be connected together to secure the amount of radiation desired.

The equivalent square feet of heating surface in one lineal foot of standard wrought iron pipe is as follows:

TABLE V

	Sq. Ft. of Surface
Diameter of pipe, ¾ in.	0.275
Diameter of pipe, 1 in.	0.346
Diameter of pipe, 1¼ in.	0.434
Diameter of pipe, 1½ in.	0.494
Diameter of pipe, 2 in.	0.622
Diameter of pipe, 2½ in.	0.753
Diameter of pipe, 3 in.	0.916
Diameter of pipe, 4 in.	1.175
Diameter of pipe, 6 in.	1.739

Heat Given off From Radiating Surface

One square foot of steam radiating surface is usually estimated to give off 250 B. t. u. per hour when operating under a pressure from 2 to 5 lb. per sq. in. in a room temperature of 70 deg. F. Assuming a steam temperature of 220 deg. which corresponds to a pressure of about 3 lb., the total difference in temperature is then 220—70 = 150 deg. 250 ÷ 150 = 1.67 B. t. u per degree difference per square foot per hour. This factor is not constant, however, and varies with the type of radiator and also the difference in temperature.

The single column radiator is more efficient than the two or three column radiator because the surface is more exposed to the surrounding air, and the air will pass over the surface more freely and produce a more rapid transmission of heat. Also the low radiator is more efficient than the high radiator for the following reason: There is a continuous upward current of air around the surface of the radiator. The air in its passage from the bottom to the top becomes heated, and as it reaches the top the transmission of heat is less rapid because of the less difference in temperature between the steam and the air.

The following table gives approximately the B. t. u. transmitted per square foot per degree difference per hour for various types of radiation when a difference of temperature is 150 deg.

TABLE VI

Type of Radiator	Height of Radiator 22 in.	26 in.	32 in.	38 in.
1 col.	1.90	1.86	1.83	1.80
2 col.	1.80	1.75	1.71	1.67
3 col.	1.70	1.65	1.60	1.54
4 col.	1.60	1.55	1.50	1.45
Window Radiator				1.85
Wall Radiator (horizontal)				1.95
Wall Radiator (vertical)				1.90
Pipe coils				2.00

To find the total B. t. u. transmitted per square foot per hour, multiply these factors by the temperature difference, which gives the following values:

TABLE VII

Type of Radiator	Height of Radiator 22 in.	26 in.	32 in.	38 in.
1 col.	285	279	275	270
2 col.	270	263	257	250
3 col.	255	248	240	231
4 col.	240	233	225	218
Window Radiator			277	
Wall Radiator (horizontal)			293	
Wall Radiator (vertical)			285	
Pipe coils			300	

The rate of transmission varies directly with the total difference in temperature between the temperature of the steam in the radiator and the room temperature. This is found to vary approximately 2 per cent for each 10 deg. variation from the standard conditions of 150 deg. difference. If the difference is less than 150 deg., the transmission will be less per degree difference than the above values, and if the difference is more than 150 deg., transmission will be higher.

Assume, for illustration, a room temperature of 60 deg. and a steam temperature of 230 deg. which gives a temperature difference of 170 deg. This is 20 deg. more than the standard conditions of 150 deg., and allowing 2 per cent for each 10 deg. variation, the values in the table giving B. t. u. per degree difference must be increased 4 per cent. Assuming a two-column radiator 32 in. high, the factor is 1.71.

$1.71 \times 0.04 = 0.068$.

$1.71 + 0.068 = 1.778$ B. t. u. per degree difference.

$1.778 \times 170 = 302$ B. t. u. per square foot per hour.

Dry Rooms

This fact is of considerable importance in estimating radiation for dry rooms or any rooms to be heated to a comparatively high temperature. Assume, for illustration, a two-column 38 in. radiator which operating normally under a temperature difference of 150 deg. will transmit 250 B. t. u. per square foot per hour. If this radiator is to be used in a dry room, in which a temperature of 140 deg. is to be maintained, what will be its B. t. u. transmission under this condition with a steam pressure of 2 lb. per square inch?

Referring to the steam tables, the steam temperature at 2 lb. pressure is 219 deg., or approximately 220 deg. The temperature difference between steam and room is $220-140=80$ deg. This is $150-80=70$ deg. below normal conditions, for which the coefficients in Table VI are given. Allowing 2 per cent for each 10 deg. variation gives 14 per cent decrease in the values given in Table VI for the above conditions

The factor for a two-column 38-in. radiator from the table is 1.67.

1.67×14 per cent $= 0.234$.

$1.67-0.234=1.436$ B. t. u. per degree difference.

1.436×80 deg. $= 114.8$ total B. t. u. per square foot emission for the radiator under the above conditions.

The total B. t. u. loss from the room per hour divided by 114.8 would give the number of square feet of radiating surface to maintain the required temperature. This method can also be used for estimating the amount of surface for any steam pressure. From the steam tables, determine the temperature of steam corresponding to the given pressure and proceed as above.

Problem

Fig. 10 shows the floor plan of a dry room to be heated to 130 deg. by means of pipe coils with a steam pressure of 2 lb. The outside wall is a 12-in brick wall and the partitions between dry room and surrounding rooms are lathed and plastered on both sides. There are two windows each 4 ft. x 6 ft. in size. Lowest temperature 0 deg., rooms surrounding dry room to be maintained at 70 deg. Room to be heated with pipe coils.

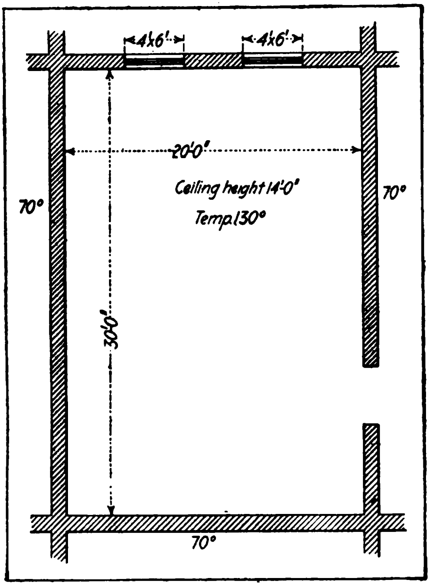


FIG. 10—PLAN OF ROOM TO SHOW APPLICATION OF RULES

Solution

$20 \times 14 = 280$ sq. ft., total outside wall and glass surface.

$4 \times 6 = 24$ sq. ft. glass exposed by each window.

$24 \times 2 = 48$ sq. ft. total glass surface.

$280 - 48 = 232$ sq. ft. net wall surface exposed.

$20 \times 30 \times 14 = 8400$ cu. ft., the total volume of room.

Applying formula, B. t. u. = $(WKw+GKg+02\ NV)\ T$.

Where Kw, the heat loss through the 12-in. brick wall = 0.30.

Kg, the heat loss through glass = 1.

G, = glass surface exposed.

N, assumed number of air changes = 1.

V, Cubic contents of room = 8400.

T, temperature of room = 130 deg.

B. t. u. = $(232\times0.3+48\times1+0.02\times1\times8400)\ 130$.

= $(69.6+48+168)\ 130$.

= $285.6\times130=37130$ lost per hour through the outside wall and air change.

To determine the loss through the partitions to the surrounding rooms $30+20+30=80\times14=1120$ sq. ft.

Kw, the heat loss through partitions $=0.36$.

130 deg. —70 deg. = 60 deg. difference in temperature.

$1120\times0.36\times60=24190$ B. t. u. per hour through partitions.

Assume a factor of 0.15 for loss through floor and ceiling.

$20\times30\times2=1200$ sq. ft. of floor and ceiling.

$1200\times0.15\times60=10800$ B. t. u. per hour.

Total loss from room $37130+24190+10800=72{,}120$ B. t. u.

The ceiling height being 14 ft. and allowing 2 per cent for each foot above 12 ft., 4 per cent should be added to the total.

$72120+2885=75{,}005$ B. t. u. per hour.

To determine the amount of 1¼-in. pipe in coils to supply the necessary heat:

Temperature of steam at 2 lb. pressure = 220 deg.

220—130 = 90 deg. difference between room temperature and steam temperature.

At 150 deg. difference in temperature pipe coils will transmit 2 B. t. u. per square foot per degree difference (from tables).

Allowing 2 per cent decrease for each 10 deg. decrease in the difference.

150—90 = 60 deg. or 12 per cent decrease.

12 per cent $\times2=0.24$.

$2-0.24=1.76$ B. t. u. per degree difference.

$1.76\times90=158.4$ B. t. u. per square foot.

$75{,}005\div158.4=474$ sq. ft of pipe.

1 ft. of 1¼-in. pipe contains 0.434 sq. ft.

$474 \div 0.434 = 1092$ ft. of 1¼-in. pipe required to heat the room.

The amount of radiator surface required may be obtained by taking the factor of heat emission given in the table, making the allowance required for the decreased difference in temperature and dividing the amount into the total B. t. u. lost.

Ratio of Volume to Square Foot of Radiation

After the amount of radiation has been determined for a given room, it is good practice to divide the volume of the room in cubic feet by the square feet of radiation necessary to determine the ratio between the two. This provides a good means of checking the computation. For average radiators with low pressure steam and room temperature of 70 deg., this ratio for various rooms should be approximately as follows:

Small rooms with two exposures 1 to 50.
Small rooms with one exposure 1 to 60.
Large rooms or lofts exposed on all sides 1 to 80.

In large office buildings, apartments, etc., where the floors are divided into small offices and rooms, the total amount of radiation on a floor divided into the entire volume of the floor which includes center halls and elevator shafts should show a ratio of 1 to 80 or 100.

These ratios will, of course, vary with different conditions, such as extremely large windows and thin walls where the ratio would be smaller or narrow rooms with comparatively small amount of exposed wall surface where the ratio would be larger. But if there is any wide variation from the above values, the computation should be checked for errors.

These values also provide a means of roughly estimating the amount of radiation in a given building to approximate the cost of the heating system, size of boilers, etc., which information is often necessary before the exact amount of radiation has been estimated.

Recessed Radiators

Radiators are often times set in recesses in walls, under shelves or enclosed behind grills, as shown in Figs. 11 to 14. Such locations decrease the efficiency of the radiators and an allowance should be made. The amount of radiation for these various conditions should be increased approximately as follows:

Fig. 11, 10%
Fig. 12, 15%
Fig. 13, 25%
Fig. 14, 30%

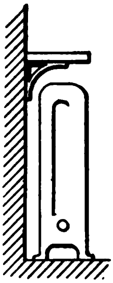


Fig. 11
Flat Shelf

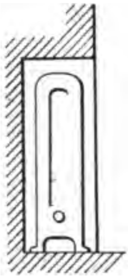


Fig. 12
In Open Recess

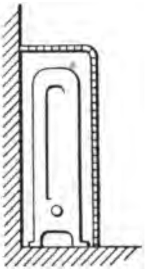


Fig. 13
Encased with
Grill

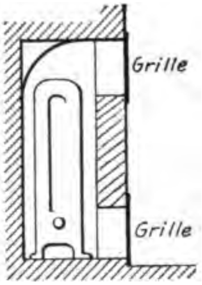


Fig. 14
Entirely Enclosed
Grill Top and
Bottom

When radiators are installed under window seats with grills at top and bottom or as illustrated in Fig. 14, the size of the grills should be such as to give at least 3 sq. in. of net free area per sq. ft. of radiating surface in the top grill and about 2 sq. in. per sq. ft. of heating surface in the lower grill.

CHAPTER VII

FLOW OF STEAM IN PIPES

THE determining factor in the flow of steam in pipes is the allowable drop in pressure between the source of steam and the most distant point to which the steam must be carried. This loss in pressure is caused by the friction of steam on the surface of the pipe and is dependent upon the velocity of flow, the size of the pipe and the length of run. Knowing these various quantities, the drop in pressure in pounds per square inch or ounces per square inch can be determined. Before taking this formula, however, the relation between the quantity and velocity will first be investigated.

Assume a pipe 1 ft. square, as shown in Fig. 15 and some fluid such as air, steam or water to be flowing through the pipe. Let two plates, A and B, be placed in the pipe perpendicular to the sides and 1 ft. apart. Assume plate B to be in the position as shown, 1 ft. from the end of the pipe. The quantity of fluid contained between the two plates will be 1 cu. ft. and the quantity of fluid between plate B and the end of the pipe would be 1 cu. ft. Assume the fluid to be flowing in the direction indicated by the arrow with a velocity of 1 ft. per second. Then in 1 sec. of time after the plates were in the position shown, each plate would have moved 1 ft. farther toward the end of the pipe, plate A moving to the position occupied by plate B and plate B moving to the end of the pipe. Therefore, 1 cu. ft. of the fluid would be discharged from the pipe per second.

If the velocity of flow is increased to 2 ft. per second, in 1 sec. of time plate A will move through a distance of 2 ft. to the end of the pipe and 2 cu. ft. of fluid will be discharged per second. It can readily be seen that if the area of the pipe is increased to 2 sq. ft. instead of 1, the quantity discharged per second would be doubled. With a velocity of 2 ft. per second and an area of 2 sq. ft., the quantity per second would be $2 \times 2 = 4$ cu. ft. We may, therefore, establish the general formula:

$Q = A \times V$ where (1)

Q = Quantity; A = Area; V = Velocity.

The formula may also be expressed as follows:

$V = Q \div A$ (2)

$A = Q \div V$ (3)

In using these formulæ it must be remembered that all quantities must be in the same units. If the velocity is given in feet per minute, the area must be expressed in square feet and the quantity will then be in cubic feet per minute.

Applying this formula to the flow of steam in pipes, two of the foregoing quantities must be known in order to determine the third. The quantity of steam is usually known or can be determined from the square feet of radiation as described in the preceding chapters, from the rate of condensation per square foot of radiation. The velocity must be assumed and the area and diameter of the pipe may then be determined.

If the quantity of steam is estimated, as stated in the foregoing, this quantity will be in pounds per hour. This must be expressed in cubic feet, and as the velocity of steam in pipes is usually given in feet per second, it should be reduced to cubic feet per second.

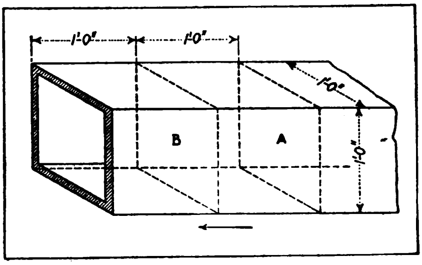


FIG. 15—DIAGRAM TO ILLUSTRATE FLOW OF STEAM IN PIPES

Referring to the steam tables given in chapter V table III the volume of 1 lb. is given in cubic feet for various pressures. Select the value corresponding to the given steam pressure, and this value multiplied by the pounds of steam per hour. This divided by 3,600 gives the cubic feet per second.

Velocity of Flow

The velocity of flow of steam in pipes should not exceed 100 ft. per second. Oftentimes this velocity is exceeded slightly, but it is not in accordance with good practice. This velocity should only be used on large pipes and should be decreased proportionately as the size of the pipe decreases. A main distributing pipe should be graded in the same direction as the flow of steam so that the water of condensation which is always present in a pipe containing steam will be carried along to some drip point. This grade should be about 1 in. in 20 ft. If it should happen that the pipe must be graded in the opposite direction the velocity should then be decreased about 50 per cent. in order to allow the condensation to flow back against the flow of steam.

With small pipes, 2 in. and under, a velocity of 20 to 30 ft. per second should not be exceeded. With vertical risers when the steam flows up, the velocity should be kept below 30 ft.

Problem

Find the size of a pipe necessary to carry 2,500 lb. of steam per hour at a velocity of 80 ft. per second, steam pressure, 10.

CHAPTER VIII

PRESSURE DROP IN STEAM MAINS

WHEN steam flows through a pipe there is a certain amount of friction caused by the steam moving in contact with the pipe which produces a drop in pressure. It is this drop in pressure which limits or determines the velocity of flow of the steam. The available drop in pressure between the boiler or the starting point of the distributing system and the most distant point must be kept within a certain proportion of the initial pressure in order that the steam will properly fill the entire system and the water of condensation be returned to the boiler.

For low pressure work from 1 or 2 lb. per sq. in. the drop in pressure should never exceed ¼ lb. at the most distant point. The standard adopted by the American Society of Heating and Ventilating Engineers is one ounce drop per 100 feet of run. On higher pressures the total drop should not exceed 1 per cent of the initial pressure. For this reason with systems having very long runs, with radiators and coils long distances from the boiler, larger mains and a lower velocity of flow must be used than in systems where the heating units are near the boiler as the drop in pressure increases as the square of the velocity. The formula derived by Professor Unwin is extensively used in determining the drop in pressure and is stated as follows:

$$P = 0.000131 \left(1 + \frac{3.6}{d}\right) \frac{W^2 l}{D\, d^5} \qquad (1)$$

Where

P = drop in pressure in pounds per square inch.

d = diameter of pipe in inches.

l = length of pipe in feet.

D = density or weight of steam per cubic foot (from steam tables).

W = pounds of steam per minute flowing through the pipe.

This formula may be transformed as follows to determine the number of pounds of steam that will be carried by any given size of pipe with an assumed drop in pressure:

$$W = 87 \sqrt{\frac{P D d^5}{\left(1+\frac{3.6}{d}\right) l}} \qquad (2)$$

The accompanying table compiled by James A. Donnelly from the Unwin formula will be found very serviceable in determining the velocity of flow and the drop in pressure in any commercial size of pipe when the amount of radiation is known. The table is based on atmospheric pressure and the rate of condensation is taken at 0.3 lb. per square foot of radiation per hour.

The first vertical column gives the nominal size of pipes. The top horizontal column gives velocities in feet per second from 10 to 130.

Below these is given the square feet of radiation that will be supplied at these various velocities, and the necessary size of pipe. Adjacent to the square feet of radiation is given the drop in pressure in ounces per square inch for each 100 ft. of pipe under the various conditions. For lengths other than 100 ft. multiply the drop in pressure by the ratio of the given length to 100. The last column gives the amount of radiation in square feet and the corresponding velocities for the various sizes of pipes if the drop in pressure is taken at one ounce per 100 ft. run.

This table applies only to steam at atmospheric pressure and would not prove correct for higher pressures. For practical purposes, however, it may be used to two pounds pressure per square inch.

The Unwin formula as given above will be found rather complicated to use for specific cases and the computation involved may be simplified considerably as follows:

Taking formula (2)

$$W = 87 \sqrt{\left(\frac{P D d^5}{1+\frac{3.6}{d}}\right) l}$$

TABLE VIII

Showing Loss of Pressure in Ounces for 100-ft. Run and Condensation 0.3 lb. Per Square Foot Per Hour Calculated from Unwin Formula

Velocity feet per sec.	10 Ft.		20 Ft.		30 Ft.		40 Ft.		50 Ft.		60 Ft.		70 Ft.		80 Ft.		90 Ft.		100 Ft.		100 ft. Run 1 oz. drop	
Size of Main	oz. Drop	Sq. Ft. Rad'n.	oz. Drop	Sq Ft Rad'n.	oz. Drop	Sq. Ft. Rad'n.	oz. Drop	Sq. Ft. Rad'n.	oz. Drop	Sq. Ft. Rad'n.	oz. Drop	Sq. Ft. Rad'n.	oz. Drop	Sq. Ft. Rad'n.	oz. Drop	Sq. Ft. Rad n.	oz. Drop	Sq. Ft. Rad'n.	oz. Drop	Sq. Ft. Rad'n.	Vel.	Rad'n. Sq. Ft.
1 inch	.4	25	1.6	49	3.6	74	6	100	10	124	14.4	148	19.6	174	25.6	198	32.4	224	40.	248	15.8	40
1¼ ins.	.29	39	1.16	77	2.61	116	6.64	155	7.25	194	10.44	232	14.21	271	18.56	310	23.5	350	29.	388	18.5	75
1½ "	.2	56	.8	112	1.80	168	3.2	224	5	280	7.2	336	9.8	392	12.8	448	16.2	504	20.	560	22.5	126
2 "	..	..	.48	200	1.08	300	1.92	400	3.	500	4.32	600	5.88	700	7.68	800	9.72	900	12.	1,000	29.	286
2½ "	..	..	..	..	.75	465	1.34	620	2.1	775	3.02	930	4.1	1,085	5.37	1,240	6.8	1,395	8.4	1,550	34.5	535
3 "	..	..	..	..	.56	669	1.	892	1.56	1,115	2.25	1,338	3.06	1,565	4.	1,784	5.06	2,007	6.25	2,230	40.	890
3½ "	..	..	..	..	..	..	.85	1,216	1.33	1,520	1.9	1,824	2.6	2,128	3.4	2,432	4.3	2,736	5.31	3,040	43.4	1,360
4 "	..	..	..	..	..	.	.66	1,600	1.04	2,000	1.5	2,400	2.04	2,800	2.66	3,200	3.37	3,600	4.16	4,000	49.	1,950
4½ "	..	..	..	..	..	..	.53	2,005	.83	2,506	1.2	3,007	1.63	3.508	2.13	4,009	2.69	4,510	3.33	5,012	54.8	2,747
5 "	..	..	..	..	..	..	.48	2,480	.75	3,100	1.08	3,720	1.37	4,340	1.92	4,960	2.43	5,580	3.	6,200	58.	3,600
6 "	..	..	..	..	..	..	..	...	.57	4,470	.82	5,364	1.12	6.258	1.47	7,152	1.86	8,046	2.3	8,940	66.	5,900
7 "	..	..	..	.	..	..	..	....	.45	6,083	.652	7.300	.887	8,516	1.16	9,733	1.46	10,950	1.81	12,167	74.3	9,040
8 "	..	..	..	..	..	..	..	....	...	....	.56	9,540	.76	11,130	1.	12,720	1.26	14,310	1.56	15,900	80.	12,700
9 "	..	..	..	..	..	..	..	....	...	...	....	....	.637	14,095	.83	16,109	1.06	18,123	1.3	20,137	87.4	17,600
10 "	..	..	..	..	..	..	..	....	...	...	...	....	.57	17,360	.75	19,840	.95	22,320	1.18	24,800	92.	22,900
12 "	..	..	..	..	..	..	..	....	...	....	....	....	....	.....	.60	28,640	.76	32,220	.94	35,800	103.	37,000
14 "	..	..	..	..	..	..	..	....	...	....	....	....	....	....	...	.....	.63	43.830	.78	48,700	113.	55,300
16 "	..	..	..	..	..	..	..	....	...	...	...	....	...	.....	...	.....	.53	57,240	.66	63,600	123.	78,300

The quantity under the radical may be divided into three separate quantities, placing a radical sign over each quantity, and the value of the formula is not changed. Therefore, this formula may be expressed as:

$$W = \left(87\sqrt{\frac{P}{l}}\right)\left(\sqrt{\frac{d^5}{1+\frac{3.6}{d}}}\right) \quad (\sqrt{D}) \qquad (3)$$

This formula is the same as the original formula, only it is expressed in a different form. For any specific case each of the three quantities in the parenthesis could be solved separately and the product of these three quantities would represent the amount of steam flowing per minute.

It will be noted that the first quantity, namely $87\sqrt{\frac{P}{l}}$, contains the drop in pressure in pounds and the length of run in feet. Let a run of 100 ft. be assumed and this quantity can then be solved for various drops.

The second quantity, $\sqrt{\frac{d^5}{1+\frac{3.6}{d}}}$ contains only the diameter of pipe as a variable. This can therefore be solved for all sizes of pipe that may be used. The third quantity $\sqrt{D}$ is the square root of the density or weight of steam per cubic foot which can be determined directly from the steam tables.

Table No. 9, compiled by W. L. Durand, gives the value of these three quantities for nearly all conditions that will be encountered in the ordinary practice. The first quantity is solved for a run of 100 ft. with the drop in pressure varying from 1 oz. to 3 lb. per square inch, and the valves are given in Column No. 1. The second quantity is solved for diameters of pipe from ½ to 12 in. the valves are given in Column No. 2. The third quantity is solved for steam pressures varying from atmosphere to 200 lb. per square inch, valves being given in No. 3.

Problem

From the following table determine the number of pounds of steam per minute that will flow through an 8-in. pipe 200 ft.

long with a steam pressure at the boiler of 60 lb. per square inch and a pressure at the extreme end of the line of 59 lb.

TABLE IX

Showing Values of Quantities in Formula (3)

COLUMN 1		COLUMN 2		COLUMN 3	
Drop in Press.	$87\sqrt{\frac{P}{100}}$	Dia. Pipe	$\sqrt{\frac{d^5}{1+\frac{3.6}{d}}}$	Steam Press. lb. per sq. in.	$\sqrt{D}$
1 oz	2.17	½ in.	0.116	0	0.193
0.1 lb.	2.75	¾ in.	0.262	2	0.207
0.2 lb	3.98	1 in.	0.522	5	0.223
0.3 lb.	4.76	1¼ in.	1.170	10	0.248
0.4 lb.	5.50	1½ in.	1.830	15	1.270
0.5 lb.	6.15	2 in.	3.670	20	0.290
0.6 lb.	6.73	2½ in.	6.050	30	0.326
0.7 lb	7.27	3 in.	11.100	40	0.358
0.8 lb.	7.78	3½ in.	16.800	50	0.388
0.9 lb.	8.26	4 in.	23.500	60	0.415
1 lb.	8.70	4½ in.	32.100	75	0.452
1.25 lb.	9.74	5 in.	43.500	100	0.508
1.5 lb.	10.60	6 in.	71.600	125	0.557
1.75 lb.	11.50	7 in.	106.000	150	0.603
2.0 lb.	12.30	8 in.	150.000	175	0.645
2.5 lb.	13.70	10 in.	272.000	200	0.684
3.0 lb.	15.00	12 in.	437.000		

Solution

The total drop in pressure for the 200 ft. is 1 lb., or a drop of 0.5 lb. per 100 ft. From Column 1, for a drop of 0.5 lb. per 100 ft. $87\sqrt{\frac{P}{l}}=6.15$. From Column 2, for an 8-in. pipe

$$\sqrt{\frac{D^5}{1+\frac{3.6}{d}}} = 150.$$

From Column 3, for steam pressure of 60 lb. $\sqrt{D}=0.415$.

From formula 3, the weight of steam equals the product of these, three quantities therefore, $W=6.15\times150\times0.415=383$ lb. per minute.

CHAPTER IX

GRAVITY SYSTEMS

ALL systems in which the water of condensation flows back to the boiler by gravity alone are called gravity return systems. In these systems the air in the radiators and pipes which is always present when a system is started or a radiator turned on after having been shut off, is removed by automatic air valves placed on each radiator and coil. When the radiator is cold the air valve is open and the air is forced out by the pressure of the steam. When the steam reaches the valve the air opening is closed automatically by expansion due to the heat in the steam, thus preventing the escape of steam.

Wet and Dry Returns

All the horizontal pipes carrying steam to the various parts of the building are called supply mains. All horizontal pipes carrying water of condensation back to the boiler are called return mains. Return mains are divided into two classes, wet returns and dry returns. Wet returns are those which are below the water line of the boiler and consequently are at all times entirely filled with water. Dry returns are those which are above the water line of the boiler and consequently are only partly filled with water and carry both water and steam. Dry returns must always be larger than is actually necessary to carry only the water of condensation. The reason for this will be explained later on.

Vertical pipes carrying steam from the supply mains up through the building to the radiators on the different floors are called supply risers. Vertical pipes carrying the water of condensation from the radiators down to the return mains are called return risers.

In the cheaper class of buildings the supply and return risers are usually run exposed in the rooms and may or may not be covered with non-conducting heat covering as the case may call for. If the risers are left uncovered they should be figured as heating surface. In the better class of buildings the risers are usually run in chases in the walls and furred in. In this case they should be covered with a non-conducting material. This is usually required by the building department in cities. In residences of frame construction the risers are usually run between the studding.

Radiator Runouts

The horizontal pipes connecting the radiators with the supply and return risers are called radiator runouts. There are three methods of installing runouts, namely, above the floor, on the ceiling and in the floor construction. The first two mentioned are objectionable because of appearance. The third method naturally makes a much neater and better appearing installation, but it is objectionable because of the difficulty in repairing these lines should leaks occur. If the building has wooden floor beams, the beams are cut to the proper size and depth to receive the pipes. In buildings of concrete construction, with steel floor beams, the pipes are usually run in the concrete floor finish above the beams. This can be done when there is 4 in. or more finish above the floor beams, but with less than 4 in. it cannot be done satisfactorily.

In this form of construction the rough concrete filling is laid in first to the level of the floor beams. The runouts from the risers are then installed, covered with non-conductive covering, and the ends capped temporarily. Over the runouts are then placed sheet metal covers bent to a half circle. The finished flooring is then laid. The sheet metal covers should be constructed of about 12 gauge black iron and of such height as to allow about 1 in. of concrete over the top.

This method of construction allows sufficient space around the pipes in the floor for expansion and contraction of risers due to the heat. The runouts should always be covered to prevent the concrete floor from cracking.

Expansion Joints

In buildings of more than 10 stories in height expansion loops should be installed in all risers to allow for expansion.

For low pressure steam the expansion in pipes will be about 1 in. in 100 ft. Expansion loops should be so constructed that all the movement is taken up by turning of the pipes in the fittings and no strain placed on any part of the piping. These loops are run close to the ceiling and usually near the floor beams so that they can be furred in. A good form of expansion loop is shown in Fig. 16.

Slip joints are sometimes used in the place of expansion loops, but it is not considered good practice to use these as they are very likely to leak and require considerable attention. If slip joints are used they should always be left exposed so as to provide access for packing.

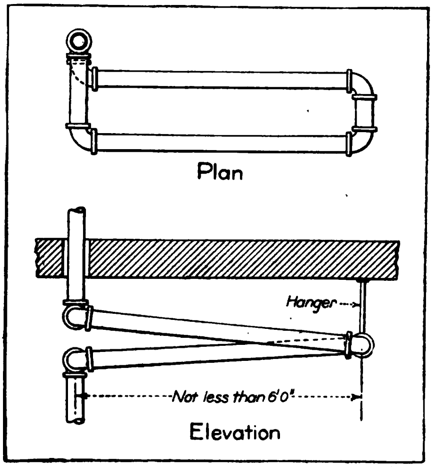


Fig 16.—Type of Expansion Loop

Single Pipe and Two Pipe Systems

Gravity return systems are divided into two general classes, namely, the two-pipe system, which has a complete system of both supply and return risers and mains and the single-pipe system with only supply mains and risers. In this system the water of condensation flows back to the boiler against the flow of steam. The single pipe system is usually designed as a combination of the two systems, the supply riser being dripped at the bottom into a return main and the water carried back separately to the boiler, as shown in Fig. 17.

Pipe Sizes for Gravity Systems

For systems of ordinary size when the runs are not over 100 ft. in length it is not necessary to estimate the drop in pressure or determine the size of mains from the velocity standpoint. It will be sufficiently accurate to size all mains and risers from table No. 10, which is for a two-pipe gravity return system. The table gives the size of supply and return mains and risers and the corresponding amount of radiation in square feet that will be supplied for a pressure of both 2 lb. and 5 lb. per square inch.

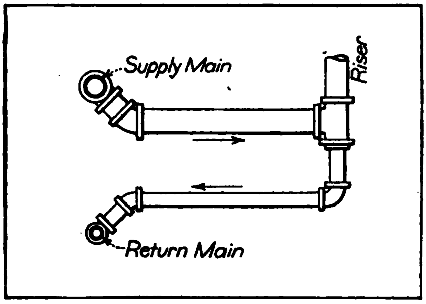


FIG. 17—DRIP FOR RISERS

TABLE X

Pipe Sizes for Two-Pipe Gravity Return System

Supply Pipe, In.	Return Pipe In.	Sq. ft. Rad. 2-lb. Press.	Sq. ft. Rad. 5-lb. Press.
1	1	40	60
1¼	1	80	120
1½	1¼	120	200
2	1½	280	500
2½	2	550	900
3	2½	900	1,500
3½	2½	1,350	2,200
4	3	1,950	3,200
4½	3	2,750	4,600
5	3½	3,700	6,200
6	3½	6,000	10,000
7	4	9,000	15,000
8	4	12,800	21,600
9	4½	17,800	30,000
10	5	23,200	39,000
12	6	37,000	62,000
14	7	54,000	92,000
16	8	76,000	130,000

Table No. XI applies to single pipe risers and mains.

TABLE XI

Single Pipe Gravity Return

Dia. of Pipe In.	Sq. ft. Rad. 2-lb. Press.	Sq. ft. Rad. 5-lb. Press.
1	30	50
1¼	70	100
1½	100	150
2	250	300
2½	500	700
3	850	1,200
3½	1,250	1,800

For a combination of the single-pipe and two-pipe systems with return mains in the basement the first table may be used for the supply and return mains.

Down Feed Systems

This term is applied to all systems in which the steam flows down the risers from the top. It is customary in this system to carry the supply mains to the ceiling of the top floor where the distributing mains are located, these in turn supplying the risers which decrease in size toward the bottom. The system is very satisfactory for single pipe work in extremely high buildings and the risers can then be sized with safety from the first table. The steam and condensation are both flowing in the same direction and much higher velocities and consequently much smaller pipe sizes can be used.

It will be noted that the sizes of the return risers and mains given in the foregoing tables are considerably larger than are actually necessary to carry the water of condensation, as the water only occupies about 1/1600 the volume of the same weight of steam at 1 lb. pressure per square inch. These apparently large sizes are necessary, however, because of the fact that steam will flow into the return risers and mains due to the condensing effect of these pipes, and if the pipes are not sufficiently large to carry this steam in addition to the water of condensation from the radiators, the steam will flow in at times at the expense of the water at various points. This action will have a tendency to hold back the water and thus interfere with the proper circulation. For this reason, wet return pipes can be much smaller than dry returns as they are at all times filled with water by virtue of their position below the water line and no steam is allowed to enter.

Location of Radiators

Radiators are usually located on the exposed wall and under windows, this being the most effective and the most convenient location. Consideration should be given, however, to the length of run from the riser to the radiator as the long runs are likely to cause poor circulation, especially in one-pipe systems. The runouts should always pitch back to the riser, otherwise a trap will be formed at the point where the connection rises to the radiator and water hammer is liable to occur. Runouts should be made amply large and it is a good practice to make these one size larger than sizes called for in the tables given for one and two-pipe gravity return systems.

Consideration should be given to the height of the first floor basement radiators above the water line of the boiler. This distance should never be less than 24 in. and should be more than this if possible. If the head room is small and it is found that the bottom of some radiators or coils are nearer to the water line than 24 in. it would be advisable to lower the boiler in a pit or adopt a boiler with a lower water line.

CHAPTER X

VACUUM SYSTEM AND VACUUM VALVES

A VACUUM system of steam circulation is a two-pipe system in which the water of condensation and the air are removed by a vacuum pump connected to the main return line. All air valves are omitted from the radiators as the air is removed with the water. The mixture is pumped by the vacuum pump to an air separating tank usually located near the ceiling of the boiler room. This tank is vented to the atmosphere. The air escapes through this vent and the water drops to the bottom of the tank. From the tank the water flows by gravity to a feed water heater. For this reason the air-separating tank must be several feet above the heater in order to overcome any pressure in the heater. A loop seal is usually installed in the connection from the tank to the heater to prevent steam from blowing out through the vent in case the tank is empty.

The accompanying illustration, Fig. 18, shows the general piping arrangement for the pumps, heaters, etc. This layout is for a high pressure plant in which the exhaust steam from the engines and pumps is used for heating. It is in this type of plant that the vacuum system is generally used, as steam can then be circulated through a very large plant with practically atmospheric pressure in the steam main, thus causing no objectionable back pressure on the engine.

With the vacuum system, some form of automatic valve is placed on the return end of each radiator and coil, the object of which is to allow the air and water to pass freely into the return main but to stop or retard the passage of steam into the return.

Types of Vacuum Valves

There are several different types of vacuum valves on the market, and these valves may be divided into three distinct classes, namely: vaporizing fluid valves, float valves and restricted orifice or weighted check valves.

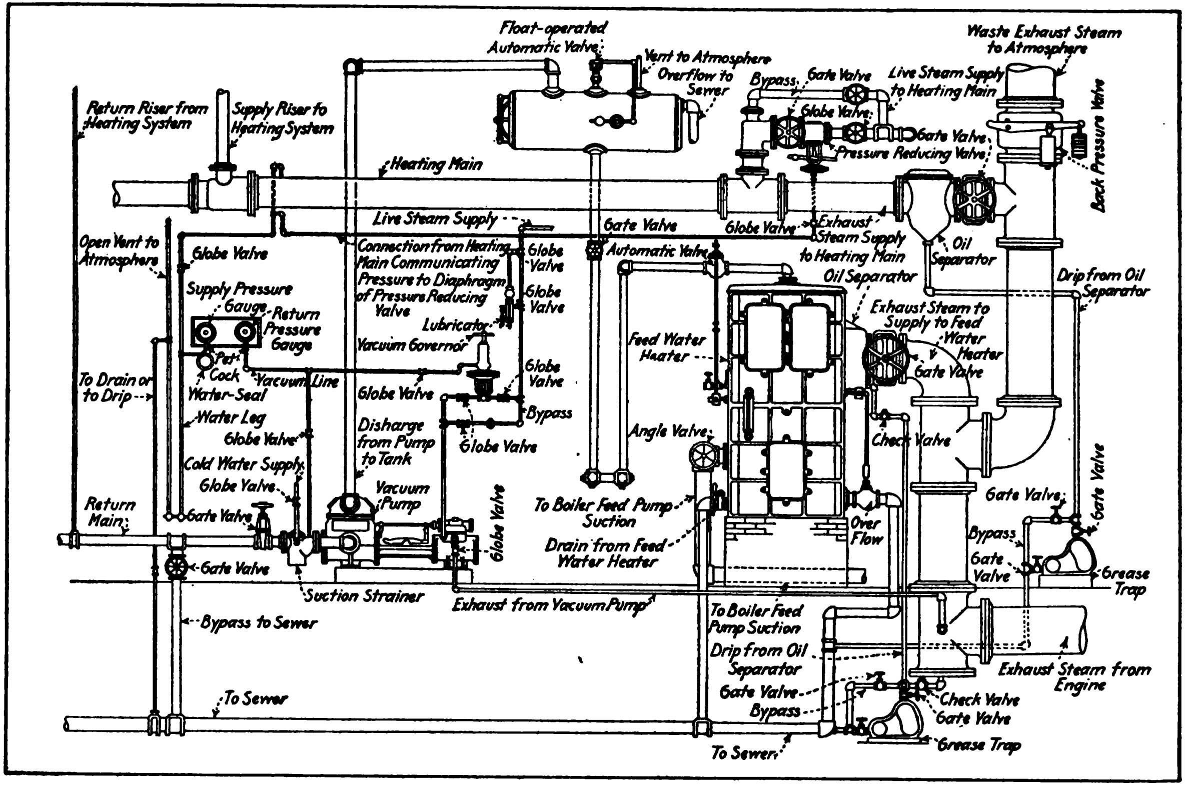


FIG. 18—LAYOUT FOR A VACUUM HEATING SYSTEM SHOWING GENERAL PIPING ARRANGEMENT FOR PUMPS, HEATERS, ETC.

The vaporizing fluid valve is a valve containing a fluid in a sealed chamber which vaporizes as soon as the steam comes in contact with the chamber. This causes an internal pressure which expands the chamber in such manner as to close the outlet of the valve, thus preventing the steam from blowing through into the return. These valves can be constructed only for constant conditions, namely, one temperature and one pressure. If the pressure and temperatures are changed the valve must have a fluid of different composition which will vaporize under the new conditions.

The float valve is constructed with a float which rises from the seat of the valve when the water accumulates and allows the water to flow into the return. This type of valve must be provided with an auxiliary passage for the air. This passage is usually through the center of the float into the return and is small in area, so that the amount of steam passing through the opening after the air has escaped is very small.

In the third type mentioned, check valves with restricted openings are placed on the return of each radiator and in addition to these, weighted check valves are placed in the base of each return riser where it enters the return main. These valves are so constructed that they can be adjusted to produce a proper flow of steam to all parts of the system regardless of the distance from the source of supply.

Fig. 18, shows the exhaust steam coming from the engine with a branch connection running to the feed water heater. The main line goes on up to the roof of the building. A back pressure valve is installed in the exhaust line above the point where the connection is taken off for the heating system. This back pressure valve is weighted so as to open when the pressure in the main exceeds about ½ lb. per square inch. If more exhaust steam is coming from the engine than is needed for the feed water heater and the heating system, the pressure will build up sufficiently to open the back pressure valve and the excess steam will escape to the atmosphere.

A pressure reducing valve is shown connecting into the main. The idea of this valve is to supply live steam direct from the boilers to the heating system at such times when exhaust steam is not sufficient to supply the demand. The valve is set to operate at a pressure slightly below the pressure at which the back pressure valve opens. If the supply of exhaust steam is not sufficient to fill the system the pressure will drop and the pressure reducing valve will open, allowing live steam to flow in until the system is filled. The pressure then builds up and the valve automatically closes.

With a vacuum system, a vacuum is maintained only in the return mains up to the return valves on the radiators. In the supply mains and radiators there is usually a slight pressure. The vacuum pump varies from 5 in. to 15 in. depending on the size of the plant. Attempts have been made to operate systems with several inches of vacuum at the source of supply but this has not proved practical.

CHAPTER XI

PIPING FOR VACUUM SYSTEMS

BOTH supply and return pipes can be considerably smaller with a vacuum system than with a gravity system, because of a greater allowable difference in pressure.

Tables 12 and 13 give the size of supply pipes and the amount of radiation that will be supplied for various length of run from 50 to 3500 ft. These tables are derived from the Unwin formula with steam at atmospheric pressure, and a total drop in pressure of 1/8 lb. and 1/4 lb. per square inch respectively.

Which of these tables to use should be determined from local conditions such as the amount of money available to spend on the system as compared with the economy, etc.

Size of Returns for Vacuum System

The return mains and connections for a vacuum system can be made much smaller than for a gravity system, not only because of the greater allowable difference in pressure, but also from the fact that a comparatively small quantity of steam will flow through the return valves into the return mains to interfere with the flow of the water. Some manufacturers of vacuum valves claim that their valves will pass absolutely no steam and these vavles may show upon testing when new practically to fulfill this claim, but after the valve has been in service for a short period of time, the accumulation of sand and scale on the seats of the valves or the escapement of the vaporizing fluid from some of the containers will cause more or less steam to pass through into the returns.

It is not a bad feature of a valve to allow a small quantity of steam into the return, if the valve is properly designed, as this will insure that the radiator will always be entirely free from air. If the valve closes absolutely tight against the passage of steam there is always a tendency in the valve to hold a small quantity of air in the radiator, which decreases the efficiency of the radiator. If a valve is so designed that all the steam flowing into the return risers is entirely condensed by the risers there can be no loss in efficiency as all the heat is utilized. As soon as it becomes necessary to condense the steam in the returns by means of jet water at the vacuum pump the system is then operating inefficiently.

TABLE XII

Square Feet of Radiation Supplied by Mains on Basis of ¼ lb. Per Square Inch Drop in Steam Pressure

Run	100	200	300	400	500	750	1000	1250	1500	2000	3000
¾	40	29	23	20	....	....	....	....	....	....	....
1	90	66	52	46	40	31	29	26	....	....	....
1¼	167	118	97	84	75	61	53	47	43	38	29
1½	277	195	154	133	123	101	87	79	72	62	54
2	609	450	375	325	295	250	210	185	175	150	133
2½	1.060	725	600	525	480	400	340	300	275	245	211
3	1,925	1,350	1,100	940	850	700	625	600	500	430	383
3½	2,750	1,950	1,575	1,360	1,250	1,000	900	800	725	625	555
4	3,850	2,725	2,225	1,910	1,725	1,400	1,250	1,100	1,025	875	775
4½	5,160	3,625	2,960	2,550	2,300	1,875	1,650	1,500	1,375	1,175	1,025
5	6,840	4,850	3,975	3,410	3,090	2,500	2,225	2,000	1,825	1,600	1,375
6	11,000	7,775	6,400	5,475	4,850	4,025	3,600	3,200	2,900	2,540	2,225
7	16,000	11,500	9,325	7,960	7,100	5,800	5,100	4,700	4,175	3,660	3,200
8	22,500	16,000	13,000	11,200	10,000	8,100	6,800	6,300	6,000	5,100	4,575
9	30,000	22,000	17,000	15,000	13,500	11,000	9,400	8,400	7,800	6,850	6,050
10	40,000	29,000	23,300	20,000	18,000	14,600	12,700	11,500	10,500	9,120	8,071
12	63,000	45,000	36,750	31,810	28,400	23,000	19,700	18,100	16,800	16,200	12,800
14	85,000	60,000	47,400	41,000	36,500	29,500	25,800	23,500	21,400	18,600	16,500
15	100,000	70,400	57,000	49,100	43,500	35,500	30,500	27,700	25,500	22,000	19,700
16	119,000	85,000	70,000	60,500	54,000	43,600	37,900	34,200	31,000	27,000	24,300
18	160,000	114,000	93,000	80,000	72,000	59,000	51,000	45,500	42,000	36,300	32,200
20	200,000	142,000	115,000	100,000	90,000	73,000	63,000	57,700	52,000	45,300	40,250

TABLE XIII

Square Feet of Radiation Supplied by Mains on Basis of ⅛ lb. Per Square Inch Drop in Steam Pressure

Run, Size, In.	100	200	300	400	500	700	1000	1250	1500
¾	28	20	16	....	....	....	....	....	....
1	63	46	36	32	28	22	20	18	....
1¼	116	83	68	59	50	43	37	33	30
1½	194	137	108	94	86	71	61	55	51
2	427	316	264	228	207	175	147	130	123
2½	745	510	425	360	338	282	240	210	143
3	1,350	950	710	660	587	482	440	420	350
3½	1,930	1,370	1,110	955	880	702	632	560	510
4.	2,700	1,900	1,560	1,340	1,210	985	880	775	720
4½	3,620	2,550	2,080	1,790	1,620	1,31	1,160	1,050	965
5	4,800	3,410	2,790	2,400	2,170	1,755	1,560	1,405	1,280
6	7,725	5,460	4,500	3,850	3,400	2,825	2,530	2,250	2,040
7	11,250	8,100	6,550	5,600	5,000	4,075	3,580	3,300	2,930
8	15,800	11,250	9,150	7,875	7,020	5,700	4,775	4.425	4,220
9	21,100	15,450	11,950	10,550	9,500	7,250	6,600	5,900	5,475
10	28,100	20,400	16.350	14,050	12,650	10,250	8,950	8,100	7,375
12	45,600	31,600	25 750	22,300	20,000	16,150	13,900	12.700	11,800
14	59,750	42,200	33,300	28,800	25,600	20,700	18,100	16,500	15,000
15	70,210	49,400	40,000	34,100	30,500	24,900	21.400	19,400	17,900
16	83,750	59,750	49,250	42,500	37,900	29,900	26,600	24,000	21,800
18	113,500	93,500	65,250	56,250	50,500	41,400	35,800	32,000	29,500
20	140,000	100,000	81,000	70,201	63,250	51,250	44,250	40,500	36,500

Table 14, compiled by J. A. Donnelly, is based on the assumption that the returns must carry the water of condensation and also the amount of steam that will be condensed by the pipes themselves.

TABLE XIV

Capacities of Return Mains and Risers under Several Conditions

No. 1 Size, Inches	No. 2 Steam Rating	No. 3 Wet Return × 40	No. 4 2½% Steam × 20	No. 5 5% Steam × 13½	No. 6 7½% Steam × 10	No. 7 10% Steam × 8	No. 8 15% Steam × 5.7	No. 9 20% Steam × 4.4
½	5	200	100	67	50	40	27	22
¾	20	800	400	270	200	160	114	88
1	40	1,600	800	540	400	320	228	176
1¼	75	3,000	1,500	1,012	750	600	427	330
1½	150	6,000	3,000	2,024	1,500	1,200	855	660
2	300	12,000	6,000	4,050	3,000	2,400	1,710	1,320
2½	500	20,000	10,000	6,750	5,000	4,000	2,850	2,200
3	900	36,000	18,000	12,150	9,000	7,000	5,130	3,960
3½	1,500	60,000	30,000	20,250	15,000	12,000	8,550	6,600
4	2,000	80,000	40,000	27,000	20,000	16,000	11,400	8,800
4½	2,800		56,000	37,800	28,000	22,400	15,060	12,320
5	3,600			48,600	36,000	28,800	20,520	15,840
6	6,000				60,000	48,000	34,200	26,400
7	9,000					72,000	51,300	39,600
8	13,000						74,100	57,200
9	18,000							79,200
10	23,000							
12	37,000							
14	55,000							
16	78,000							

The first column gives the diameter of pipe in inches. Column No. 2 gives the steam rating of the pipes based on a drop of 1 oz. per 100 ft. run. This rating is rather conservative for vacuum systems. The preceding tables are recommended for sizing the supply mains in preference to this. For sizing the return mains column No. 3 gives the amount of radiation for the corresponding pipes as wet returns carrying no steam. Columns 4 to 9 inclusive give the ratings of returns for 2½ per cent. to 20 per cent. exposed surface. Which means that in a particular system, if the amount of exposed heating surface in the return mains is 2½ per cent. of the total radiating surface in the system the return mains must carry the condensation from the entire system plus the 2½ per cent. of steam which will be condensed by its own surface. The main should then be sized from the 2½ per cent. column or No. 4.

For all ordinary vacuum systems it will be found that best results will be obtained by sizing the return risers from the 7½ per cent. column, or No. 6, and the return mains in the basement from the 2½ per cent. column, or No. 4.

Vacuum Systems for Low Pressure

The system as described in the preceding paragraphs is the type which is usually adapted for a high-pressure plant. Steam-driven vacuum and boiler-feed pumps are used and the exhaust from these pumps turned into the heating system. In buildings where steam is used for heating purposes only, this arrangement would necessitate the boilers being operated at high pressure in order to operate the pumps. This necessitates a licensed engineer to operate the heating systems which is expensive and there are various other objections so that this arrangement is not generally considered good practice. One means of obviating this is to install electric-driven boiler feed and vacuum pumps which is often done.

Vacuum Systems Without Boiler Feed Pumps

Vacuum systems are sometimes designed to operate without the use of a boiler feed pump by carrying the discharge from the vacuum pump into a standpipe. This standpipe should be not less than 6 in. in diameter and should be carried up 12 or 15 ft. above the water line of the boiler. The top of the pipe should be vented to the atmosphere and also provided with an overflow to the sewer. From the bottom of the standpipe a return line is run to the boiler. The discharge from the vacuum pump is connected into the standpipe at any convenient height, preferably about 5 ft. above the water line.

The general arrangement of the piping is shown in Fig. 19. The operation of the system with this arrangement is the same as before mentioned with the exception that the standpipe takes the place of the air separating tank and boiler feed pump.

The vacuum pump withdraws the air and water from the radiators through the return main and delivers the mixture into the standpipe. The air passes up the standpipe and out to the atmosphere through the vent. The water will rise in the standpipe until the head accumulated is sufficient to overcome the boiler pressure and then flow back by gravity to the boiler. As the pressure on the boiler is seldom more than 2 lb. per square inch the maximum height of the water in the standpipe will not exceed 5 ft.

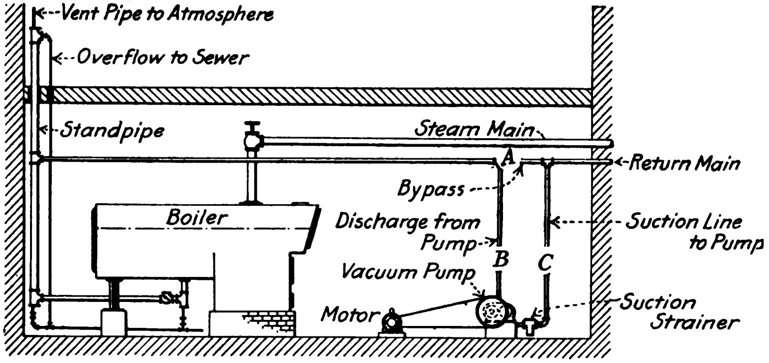


FIG. 19—GENERAL ARRANGEMENT OF PIPING FOR VACUUM SYSTEM WITHOUT BOILER FEED PUMP

A bypass with three valves A, B and C is shown around the vacuum pump, so that the system can be operated temporarily without the use of the vacuum pump. Normally valve A is closed and valves B and C are open. If it is desired to cut out the vacuum pump after the circulation is well started valves B and C are closed and valve A is opened. The water then flows directly back to the boiler by gravity and any accumulated air

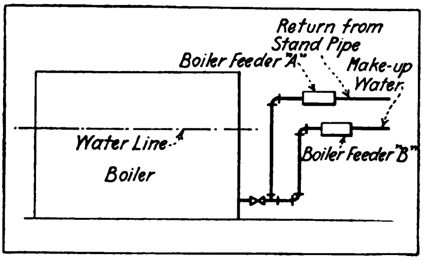


FIG. 20—ARRANGEMENT FOR TWO AUTOMATIC BOILER FEEDERS

will pass off through the vent pipe. This is only possible when the lowest point in the return is at least 4 ft. above the water line of the boiler.

If it is desired to make the system entirely automatic it is well to provide two automatic boiler feeders as shown in Fig. 20.

Boiler feeder A is installed in the return from the standpipe. Boiler feeder B is installed in the cold-water make-up line. Feeder A is so located as to control the upper limit of the water line. Should a large quantity of water be brought back suddenly from the heating system and tend to flood the boiler, feeder A will close and prevent this. If for any reason water should be held back in the returns in large enough quantities to cause the water line in the boiler to drop below the lower limit, then feeder B, which is set about 2 inches below feeder A, opens and allows sufficient fresh water to be admitted to make up the deficiency.

CHAPTER XII

VACUUM PUMPS FOR VACUUM HEATING SYSTEMS

FOR vacuum systems of steam circulation where high pressure steam is available to operate the pumps the type most generally used is the simplex double acting steam vacuum pump. It is not necessary to use a high vacuum or high duty pump as the service is very light. From five to ten inches of vacuum at the pump is sufficient for all ordinary systems, and the head against which the pump must discharge is the distance in feet between the pump valves and the highest point in the discharge line to the air separating tank. This height is usually from 15 to 20 feet.

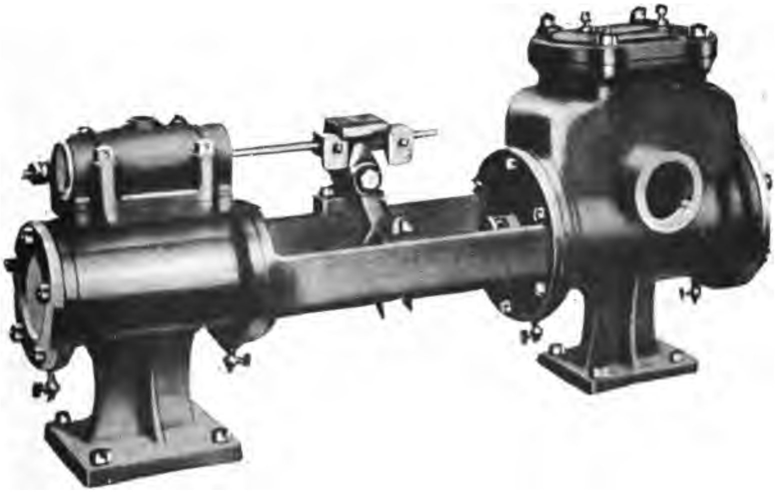


FIG. 21

Figure 21 shows a type of vacuum pump manufactured by the Union Steam Pump which is well adapted for vacuum heating work. Both the suction and discharge connections are on the side which the casing over the valves to be removed for access to the interior without disturbing the piping.

Capacity of Vacuum Pumps

As has been pointed out before, the quantity of water which must be handled by the vacuum pump is comparatively small. The heaviest service comes when steam is first turned on in the morning, after the building has been allowed to cool down. The condensation for the first half hour will be several times as much as the normal condensation, due to necessity of bringing the iron in the piping and radiators up to temperature. The pump must also have sufficient capacity to remove the air from the system quickly, in order that the circulation may be rapid.

It is therefore necessary that the pump be considerably larger than is actually required to handle the normal condensation. Table 15 gives the different sizes of the standard makes of low vacuum pumps and the estimated amount of direct radiation in square feet that they will handle. These figures are based on pumps capable of handling ten times the quantity of water condensed by the corresponding amount of direct radiation under normal conditions.

TABLE XV

Vacuum Pump Capacities

Size of Pump	Sq. ft. Rad.	Floor Space
4 x 5 x 5	4,000	9″ x 34″
4 x 6 x 7	8,000	14″ x 43″
5½ x 8 x 7	16,000	14″ x 45″
6 x 9 x 10	25,000	20″ x 58″
6 x 10 x 12	40,000	24″ x 67″
8 x 12 x 12	70,000	25″ x 67″
8 x 14 x 12	100,000	28″ x 76″
8 x 14 x 16	130,000	30″ x 82″
10 x 16 x 16	175,000	30″ x 84″
12 x 18 x 18	225,000	32″ x 88″
12 x 18 x 24	250,000	32″ x 116″
14 x 20 x 24	275,000	34″ x 118″

The size of the pumps are given in inches. The first dimension is the diameter of the steam cylinder, the second the diameter of the water cylinder and the last dimension is the length of the stroke.

The exhaust steam from these pumps should always be connected into the main exhaust line back of the oil separator and thus used in the heating system. In large plants it is always advisable to install two pumps each of which is capable of handling the plant alone. This arrangement provides an emergency pump in case of repairs.

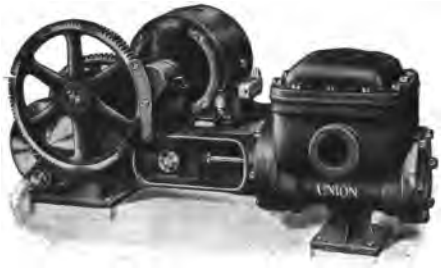


FIG. 22

When a vacuum system is used in connection with a low pressure plant an electric driven pump must be used.

Fig. 22 shows an electric driven reciprocating vacuum pump of the same type as shown in Fig. 21. The motor is mounted directly on the frame of the pump and the drive is through reduction gearing. Rawhide pinions should be used in order to eliminate noise. This type of pump is also made with separate motor base and belt drive.

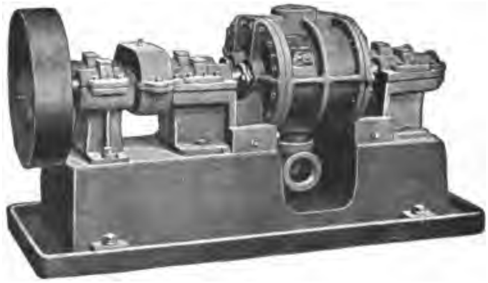


FIG. 23

Rotary Vacuum Pumps

When extremely quiet operation is desired the rotary vacuum pump is the best type to use. This pump is very simple in design having only two moving parts and therefore requires little attention. An illustration of this pump is shown in Fig. 23. The vacuum is created by the lobes moving against the casing as shown in the section, Fig. 24. The arrows indicate the direction of rotation as well as the direction of flow of the water and air passing through the pump. The lobes are so constructed that they mesh together in the center in the same manner as gears and thus prevent any communication between the suction and discharge.

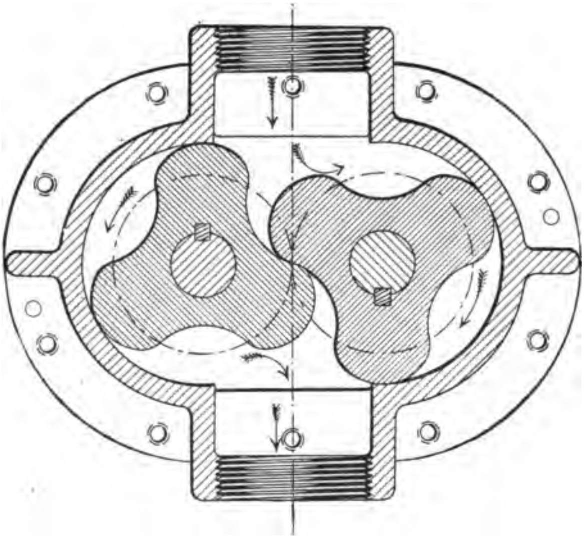


FIG. 24—

Capacity of Rotary Pumps

The sizes of rotary pumps are given in terms of the diameter and length of the rotars. These dimensions are given in inches. Table 16 gives the various sizes of the standard makes, their capacities at different speeds and the required horse-power for driving.

The figures for horse-power are based on the pumps operating at a vacuum of ten inches and discharging against a pressure of ten pounds. Usually the discharge head will be less than this and the actual power required will be considerably less than the quantities given in the table.

TABLE XVI

Capacities of Rotary Vacuum Pump

Size of Pump	Spuds R. P. M.	Sq. ft. Radiation	Horse Power
3 x 3½	300 to 500	4,000 to 6,000	¾ to 1½
4 x 4	250 " 400	6,000 " 8,000	1½ " 2½
4 x 6	250 " 400	8,000 " 16,000	2½ " 3½
5 x 8	200 " 300	16,000 " 25,000	3½ " 4½
6 x 9	200 " 250	25,000 " 50,000	4½ " 6½
8 x 8	150 " 225	50,000 " 70,000	6½ " 10
10 x 10	125 " 200	70,000 " 130,000	10 " 14

Both the suction line and discharge from these pumps should be provided with compression air chambers. The construction of these chambers is shown in Fig. 25. This drawing also shows

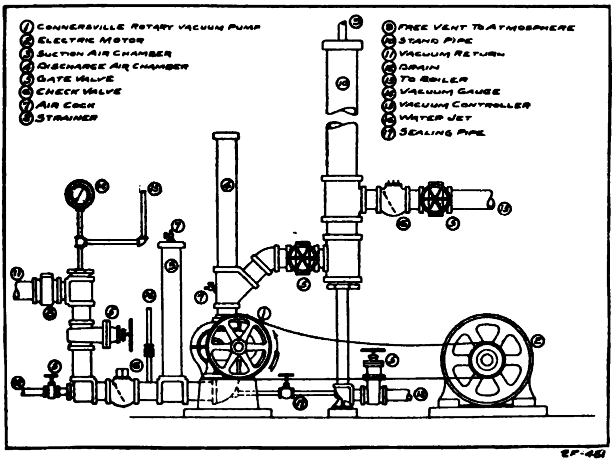


FIG. 25

the method of making the pump connections when a stand pipe is used to feed the water to the boiler as previously described.

CHAPTER XIII

INDIRECT SYSTEMS

AS has been stated in Chapter V, indirect systems or more properly gravity-indirect systems are those in which the radiator is placed in a galvanized iron box below the room to be heated and the heated air carried through ducts to the room. The term "gravity indirect" is used because the air flows by virtue of the force of gravity over the heater and into the room. This is because the air immediately surrounding the radiator or the indirect stack is heated and expanded. This air being lighter than the colder air around it is forced upward by the heavier column of colder outside air and thus enters the room. The cold air taking its place is in turn heated and circulation is thus established.

There are several kinds of extended-surface indirect radiators manufactured and rated in various catalogues. The ratings given are, in some cases, rather uncertain. The best results will be obtained by the use of "Vento" radiation which has been very carefully tested under various conditions and the ratings are conservative. Vento radiation is discussed more fully in Chapter XXI.

The estimating of the amount of indirect surface necessary is rather uncertain as various conditions enter in to influence the results obtained, such as the horizontal length of the ducts, the directness of the passage of the air, etc. These several points should be considered in making the design. In order to have the system operate satisfactorily a vent or some means of allowing the air in the room to escape should always be provided. If the room is provided with a fire-place, as is often the case, this forms a very good vent. The warm-air inlet should be so located in reference to the vent opening as to form a proper distribution of the air.

The air entering the room is always at a considerably higher temperature than the desired room temperature. It is the cooling of this heated air down to the room temperature that supplies the heat lost by conduction and radiation from the walls and windows. It can readily be seen then that the B. t. u. given up by the quantity of air entering the room in any given period of time, cooling from the entering temperature down to 70 deg. must equal the B. t. u. lost from the walls and windows (not including the air change) in the same period of time to heat the room.

To estimate the amount of radiation necessary the heat loss from walls and windows as previously explained must first be determined. Using the same terms as before.

W = Net exposed wall in square feet.
K_w = Factor of heat transmission for wall exposure.
G = Total glass exposure in square feet.
K_g = Factor of heat transmission for glass exposure.
T = Room temperature, degrees Fahrenheit.
T_o = Lowest outside temperature, degrees Fahrenheit.
T_1 = Temperature of air entering room, degrees Fahrenheit.
Q = Cubic feet of air entering room per hour.
H = B. t. u. loss from walls and windows (not including air change).

Then

$$W \times K_w + G \times K_g = H \qquad (1)$$

In this formula should also be included any other losses that may occur such as floor losses, ceiling losses, etc. Percentages for north and west exposures, high ceilings, etc., should also be added to this.

One B. t. u. will raise 55 cu. ft. of air one degree. Therefore 55 cu. ft. of air cooling one degree will give up one B. t. u. Or one cu. ft. of air cooling one degree will give up $\frac{1}{55}$ B. t. u. If there are Q cu. ft. of air entering the room per hour and cooling from a temperature of T_1 down to T the amount of heat given up will be $\frac{Q}{55}(T_1-T)$ = B.t.u. given up per hour. (2)

As this must equal the heat lost per hour, equation 1 must equal equation 2. Therefore

$$\frac{Q}{55}(T_1-T) = H \qquad (3)$$

Solving this for Q we have

$$Q = \frac{H \times 55}{T_1 - T} \qquad (4)$$

This gives the cubic feet of air that must be admitted to the room per hour.

The temperature of the air entering through the register (T_1), under ordinary conditions, is about 120 deg., and with a room temperature of 70 deg. formula (4) may be reduced to

$$Q = \frac{H \times 55}{50} \qquad (5)$$

This air must be heated through a range of ($T_1 - T_o$) or usually 120 deg. by the indirect stack. Knowing the quantity of air and the range through which it must be heated the number of B. t. u. and consequently the size of the stack can be determined. Taking formula (3) and inserting T_o in place of T we have

$$\frac{Q}{55}(T_1 - T_o) = H_1 \qquad (6)$$

Where H_1 is the B. t. u. that must be supplied by the indirect stack.

An indirect stack will transmit considerably more heat units per square foot of surface than a direct radiator, because the air passes over the surface at a higher velocity and is at a lower temperature when coming in contact with the surface. One square foot of surface will transmit about 350 B. t. u., and this value may be used under ordinary conditions with safety.

We may therefore write the final formula.

$$\frac{H_1}{350} = \text{Sq. ft. of surface necessary.}$$

The amount of surface thus found should be increased from 10 per cent. to 15 per cent. for wind leakage.

To determine the size of ducts from the stack to the rooms the following velocities should be used.

First floor:	200 ft. per minute.
Second floor:	300 ft. per minute.
Third floor:	400 ft. per minute.

Applying formula $A = \frac{Q}{V}$ we have for the size of ducts in square inches.

$$\text{First floor } A = \frac{Q \times 144}{200}$$

$$\text{Second floor } A = \frac{Q \times 144}{300}$$

$$\text{Third floor } A = \frac{Q \times 144}{400}$$

The area of cold-air ducts are made about 80 per cent. of the area of the warm-air ducts. The area of the register should be about 40 per cent. larger than the area of duct.

The general arrangement of the duct work around the heater is shown in Fig. 26.

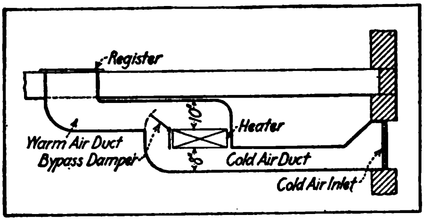


FIG. 26—CONNECTION TO INDIRECT RADIATOR

The by-pass shown is for allowing a portion of the air to pass around the heater and thus provide a means of controlling the temperature in moderate weather. This damper may be operated by a thermostat placed in the room and thus make the temperature control automatic. The amount of indirect surface may be estimated approximately by adding 50 per cent. to the amount of direct surface necessary for the old style pin or extended surface radiators. For vento radiation the amount of surface necessary should be about the same as the amount of direct radiation.

The amount of direct-indirect is usually estimated by adding 25 per cent. to the amount of direct surface necessary.

To determine the size of main and boiler capacity for indirect systems it is necessary to reduce the amount of radiation to the equivalent direct radiation. If Vento radiation is used the actual amount of surface multiplied by three will give the equivalent amount of direct radiation.

CHAPTER XIV

BOILERS

FOR residence work and comparatively small buildings cast-iron boilers may be used for heating purposes with both steam and hot-water systems and are very satisfactory. For buildings containing 5000 or more square feet of radiation steel boilers of the brick set return tubular type or the fire-box return tubular type give better satisfaction and are more economical than the cast-iron type.

The ratings of cast-iron boilers are given in the manufacturers' catalogues. These ratings are given in square feet of radiation, meaning the amount of direct radiation the boiler will supply. These boilers are, as a rule, considerably overrated and allowance for this should be made in selecting the size of the boiler to be used. An allowance should also be made for the heat losses from steam and return and risers. When all pipes are covered with non-conducting covering, 50 per cent. should be added to the total amount of direct radiation and a boiler of this capacity used. This allows for the loss in the piping and the overrating of the boilers by the makers and a certain amount for reserve capacity. If all piping is left exposed the total amount of surface in the piping should be figured and added to the direct radiation. This amount should then be increased about 40 per cent. for determining the size of the boiler.

Boiler Horse Power

One boiler h. p. is the work or heat necessary to evaporate 34½ lbs. of water per hour from a temperature of 212 deg. into steam at 212 deg. F., or at atmospheric pressure.

Under normal conditions each square foot of radiating surface will condense about 0.25 lb. of steam per hour. Therefore, one boiler horse-power will supply $\frac{34.5}{0.25} = 138$ sq. ft. of radiating surface. It is usually estimated that one boiler horsepower will supply 100 sq. ft. of surface. This is, however, an unsatisfactory way to determine or specify the size of a boiler as the term horse power is not definite enough.

To illustrate, a boiler with a comparatively small amount of heating surface might fulfill the above requirements of evaporating 34½ lb. of water per hour per rated horse-power, but it might be necessary to force the boiler to do so. This would necessitate the burning of an excessive amount of coal and consequently a loss in efficiency. A much more satisfactory and definite method to specify the size of a boiler is to state the actual amount of heating surface which the boiler must contain.

Boiler Heating Surface

The term heating surface as applied to a boiler means the surface of the boiler, which is in contact with the fire or hot gases on one side and water on the other. Any portion of the boiler which is in contact with the hot gases on one side and steam space on the other is called superheating surface and should not be considered as actual heating surface. This latter condition, however, is very seldom met with in any standard boilers.

To determine the square feet of heating surface it is necessary to fix the rate of evaporation, or the number of pounds of steam that will be generated per square foot of heating surface per hour. It has been determined through experience with boilers ranging from 50 h. p. to 125 h. p. that an evaporation of about 3 lb. of water per square foot of heating surface gives the best results. For boilers ranging from 35 h. p. to 50 h. p. an evaporation of 2½ lb. and below 35 h. p. an evaporation of 2 lb. should be assumed. From the above assumptions the following general rules may be established: When the total quantity of steam to be supplied is less than 900 lb. per hour, then:

$$\frac{\text{Pounds of steam per hour}}{2} = \text{Boiler heating surface}$$

Between 900 lb. and 1250 lb.:

$$\frac{\text{Pounds of steam per hour}}{2\frac{1}{2}} = \text{Boiler heating surface}$$

Above 1250 lb.:

$$\frac{\text{Pounds of steam per hour}}{3} = \text{Boiler heating surface.}$$

The rules already given in the preceding chapters should be followed in determining the total quantity of steam per hour. To summarize these again, first determine the condensation of steam per square foot of radiating surface under the various conditions. These rates multiplied by the corresponding amount of exposed surface give the total quantity of steam required. If part of the radiation is indirect or direct-indirect the same method should be followed, using the figures given for these conditions.

If all steam mains, returns and risers are left uncovered all this exposed surface should be included. If pipes are covered it is well to add from 10 per cent. to 15 per cent. to the total for losses through covering around the boilers, etc.

Boiler Grate Surface

The proper size or proportion of the grate surface of a given boiler should be as carefully considered as the size of the boiler itself, and the dimensions should always be given in connection with the boiler.

To determine the size of a grate for any given boiler we must know what ratio of grate surface to heating surface to assume and this ratio depends upon two things, namely, the rate of evaporation or the number of pounds of water that will be evaporated per pound of coal burned and the number of pounds of coal that will be burned per square foot of grate.

Boiler Efficiency

The first of these two depends upon the efficiency of the boiler and the quality or number of heat units in the coal. The term "boiler efficiency" means the ratio of the heat put into the boiler or the total heat in the coal burned to the heat given out or available in the steam as it leaves the boiler. The chief source of loss is in the flue gases which leave the boiler at a comparatively high temperature and also radiation of heat from the boiler walls.

The efficiency of a boiler will vary from 50 per cent. to 75 per cent., depending upon the size and type, the setting, the conditions of the draft, method of firing, etc. In some cases the efficiency may be somewhat higher than 75 per cent. in very large and carefully managed plants. On the other hand, the efficiency may fall considerably below 50 per cent. in small plants where less careful attention is paid to the operation of the boilers.

Rate of Evaporation

Assuming a boiler efficiency of 60 per cent. and the heating value of coal at 13,000 B. t. u. per lb., $13{,}000 \times 0.60 = 7800$ B. t. u. available per pound of coal. With feed water entering the boiler at 212 deg. F. and evaporated into steam at atmospheric pressure we find upon referring to the steam tables that this requires 966 B. t. u. per pound of water evaporated (latent heat of steam under the above conditions). Therefore $7800 \div 966 = 8.0745$ lb. of water evaporated for each pound of coal burned. For average conditions with boilers of 50 h. p. and over an evaporation of 8 lb. of water per pound of coal may safely be used. For boilers of 35 to 50 h. p. capacity 7 lb. of water per pound of coal and boilers below 35 h. p. 6 lb. of water per pound of coal.

Combustion per Square Foot of Grate

The rate of combustion of coal depends primarily upon the strength of the draft and upon the grade or quality of the coal. As the strength of draft varies approximately as the square root of the height of the chimney for natural draft, this should be considered in estimating the rate of combustion. As to the quality of coal, bituminous coal is the most rapid burning; next comes the semi-bituminous, the semi-anthracite and last the anthracite. Also, the smaller the size of the anthracite the less rapid will be the burning. For ordinary house heating boilers where the firing is more or less irregular the rate of combustion will probably not exceed 6 lb. per square foot per hour. With larger power boilers the rate may be from 10 to 20 lb.,depending upon the conditions mentioned above. With forced draft or very high stacks the rate may be considerably more than this, but the higher rates are apt to affect the economy of operation.

Referring now to the rates of evaporation or the pounds of steam generated per pound of coal, namely, 6 lb., 7 lb. and 8 lb. for the different sizes of boilers, if the total quantity of steam to be supplied is divided by the rate of evaporation the result will be the quantity of coal that must be burned per hour to produce the necessary steam. The total quantity of coal per hour divided by the assumed rate of combustion per square foot per hour will give the necessary grate surface. We may, therefore, establish the following general rules for determining the grate surface:

When the amount of steam to be supplied is less than 900 lb. per hour:

$$\text{G. S.}=\frac{\text{Pounds of steam per hour}}{6\times\text{Coal per square foot.}}$$

Between 900 and 1250 lb.:

$$\text{G. S.}=\frac{\text{Pounds of steam per hour}}{7\times\text{Coal per square foot.}}$$

Above 1250 lb.

$$\text{G. S.}=\frac{\text{Pounds of steam per hour}}{8\times\text{Coal per square foot.}}$$

For the first case the pounds of coal per square foot should be assumed from 6 to 12; for the second case, from 8 to 16, and the third case from 10 to 20, depending upon draft conditions, etc., as mentioned above.

Problem

Assume a building containing 12,000 sq. ft. of radiating surface composed of coils, 5000 sq. ft. of which are in storage rooms which are maintained at a temperature of 50 deg. F. The remainder is in rooms heated to 70 deg. F. In addition to the above there are 500 sq. ft. of indirect radiating surface. Find the actual size of boiler and grate surface necessary to available steam pressure of 3 lb.

Solution

Referring to the tables of B. t. u. transmission from radiating surfaces given in Chapter VI, Table VI it is found that for pipe coils with 150 deg. difference between steam temperature and room temperature the factor is 2, or $2 \times 150 = 300$ B. t. u. per sq. ft. in the rooms heated to 70 deg. The B. t. u.given up per pound of steam is 1000, therefore, $300 \div 1000 = 0.3$ pound of steam condensed per square foot.

For 7000 sq. ft. of radiation, $7000 \times 0.3 = 2100$ lb. of steam required per hour.

The transmission from the coils in the storage room will be 4 per cent. more or 2.08 B. t. u. per degree difference, as the difference in temperature between the steam and the room is 170 deg. instead of 150 deg. Total B. t. u. transmission is $170 \times 2.08 = 354$ B. t. u.

Steam condensed per square foot, $354 \div 1000 = 0.354$ lb.

For 5000 sq. ft. of radiation, $5000 \times 0.354 = 1770$ lb. per hour.

Steam condensed by indirect surface, $\frac{350}{1000} \times 500 = 175$ lb.

Total steam required, $2100 + 1770 + 175 = 4045$ lb.

Adding 10 per cent. for losses through mains, etc., $4045 + 405 = 4450$ lb. of steam per hour.

$\frac{4450}{3} = 1483$ sq. ft. of heating surface necessary in the boiler. With a rating of 12½ sq. ft. of heating surface to one boiler horse-power, which is the usual rating, this would require actually 118.7 horse-power or approximately 120 horse-power.

To determine the grate surface necessary the height of stack, grade of coal, etc., should be known. Assuming average conditions with 15 lb. of coal per square foot of grate, then:

$\frac{4450}{8 \times 15} = 37$ sq. ft. grate surface.

Types of Boilers

In large buildings where heating alone is required and no high pressure steam is necessary for running engines or for manufacturing purposes, the brick set horizontal return tubular and the firebox return tubular boilers, shown in Figs. 27 and 29 respectively, are very satisfactory and also economical.

The firebox type, shown in Fig. 29, is used very extensively for school work. The fire being entirely within the boiler and surrounded by a water jacket reduces the radiation losses to a

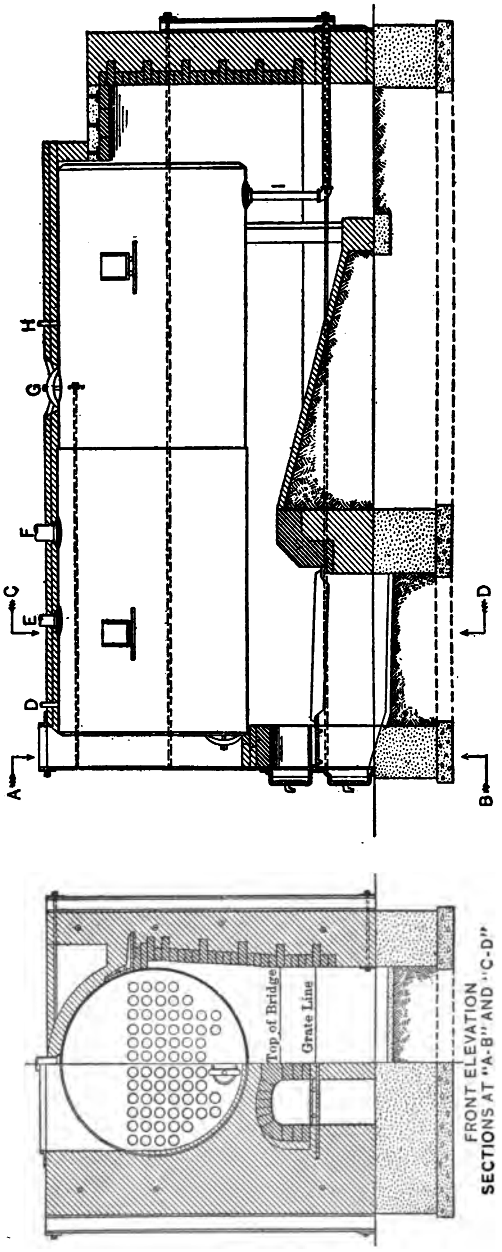


FIG. 27—FRONT ELEVATION AND SECTION THROUGH BRICK-SET HORIZONTAL RETURN TUBULAR BOILER

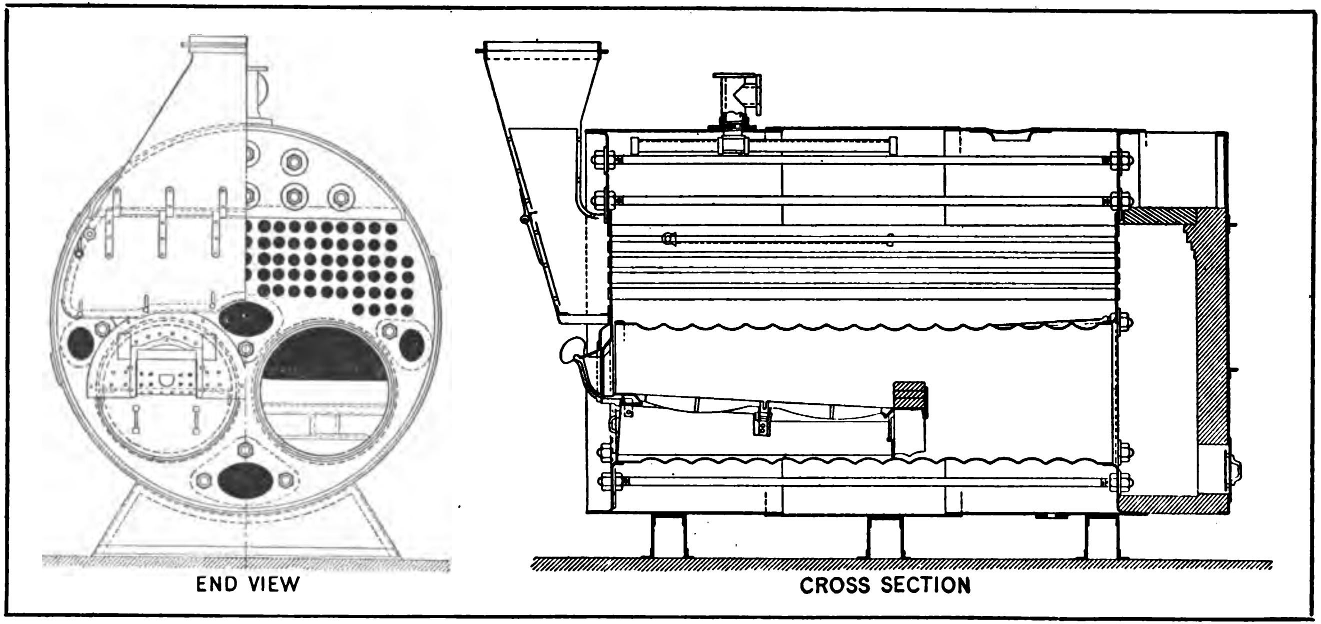


FIG. 28—END AND SECTIONAL VIEW OF SCATCH MARINE TYPE BOILER

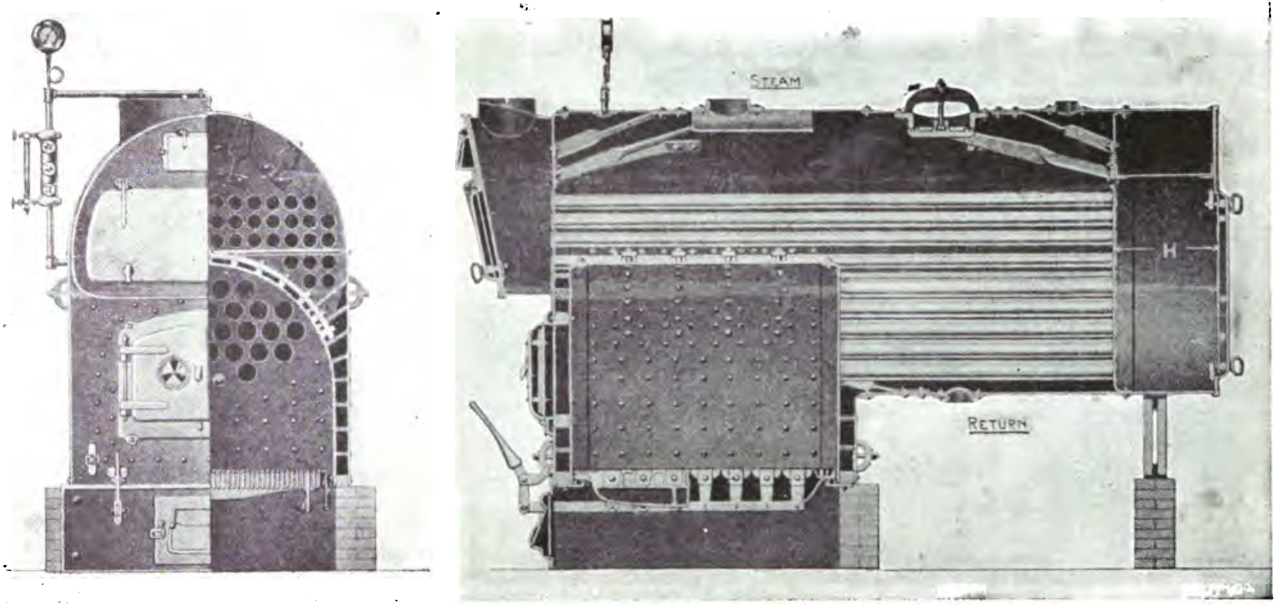


FIG. 29.—TYPE OF HORIZONTAL TUBULAR BOILER USED EXTENSIVELY IN SCHOOLHOUSE HEATING

minimum. The gases have a comparatively long travel before reaching the flue and are therefore reduced to their minimum temperature and for this reason the boiler is very efficient. No brick setting is required with the exception of that at the ashpit, which is usually built of brick, though in the smaller sizes it may be constructed of cast iron. The exposed surface of the boiler should be covered with nonconducting covering. Asbestos sectional blocks 1½ in. thick, wired on and finished with hard cement forms a good covering. The boiler is also used to some extent for high pressure and power purposes, though seldom where pressures above 100 lb. per square inch are required.

The return tubular boiler is used a little more extensively for power purposes than the firebox type up to 100 lb. pressure per square inch and in units up to about 150 h. p. The difficulty encountered in constructing these boilers for higher pressures and large units is that the boiler shell must be so thick to withstand the pressure that the heat from the fire is not readily transmitted to the water on the inside without injuring the comparatively thick metal of the shell. With the Scotch-marine type of boiler shown in Fig. 28, this difficulty is eliminated because the fire is entirely within the shell, and the shell may be made any thickness that is practicable. The furnaces in the Scotch-marine boilers are constructed with corrugations, as shown in the illustration. This enables the furnace to withstand many times the pressure carried with the same thickness of metal that it would withstand without the corrugations because the pressure is external and tends to crush the cylinder. This type of boiler is very efficient because the furnace is entirely surrounded by a water jacket and the radiation losses are comparatively small.

All of the above boilers are classified as "the fire tube type" because the fire is on the inside of the tubes. The other general class of high pressure boilers is called the "water tube type," as shown in Figs. 30 and 31, in which the water is on the inside of the tubes and the fire on the outside. This type of boiler is most commonly used in up-to-date power plants. The element of danger is much less with these boilers than with the fire tube type and they are therefore generally used in office and other buildings, where the risk is greater. The operating pressure usually adopted in water tube boiler plants for power purposes is from 125 lb. to 150 lb. per square inch.

The boiler shown in Fig. 31 is of the longitudinal drum type, the drum being parallel with the tubes. Fig. 30 shows the cross drum type, the drum running at right angles to the tubes. The latter type is slightly less expensive than the longitudinal drum boiler and requires less head room. It has the disadvantage, however, of having less steam space and less area of water surface at the water line. The first of these objections causes a greater fluctuation in the water level with a sudden increase in the load.

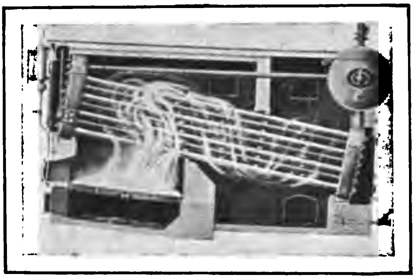


FIG. 30—WATER TUBE BOILER OF THE CROSS DRUM TYPE

The smaller water area or steam liberating surface causes more entrained moisture to be carried up with the steam as the steam must leave the water at a higher velocity. In all of the above types of boilers the square feet of heating surface can be very easily calculated, as all of the surfaces are regular and their areas can be figured directly from the plans and the manufacturers' rating checked thereby.

One of the most important things to observe in plant design is the determination of the size and number of boilers to use after the maximum load has been estimated. In ordinary residences and small buildings it is customary to install one boiler of sufficient capacity to carry the load in extreme weather conditions. In larger buildings, however, it is not safe to follow this rule. If the plant requires from 50 h. p. to 200 h. p. two units should be installed, the combined capacity of which is from 25 to 50 per cent. in excess of the total load. With this combination the load could be carried by one boiler even in extreme conditions for a short period of time in case repairs should be necessary to the other boiler. In addition to this, during average weather conditions one boiler would carry the load and operate very closely at its rated capacity. This is obviously much more economical than running one large boiler on a light load.

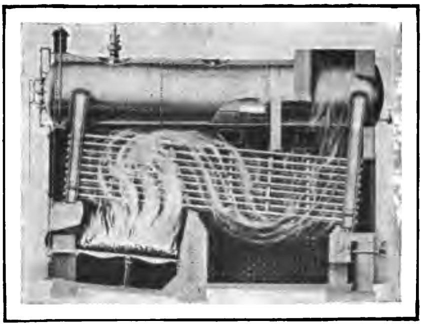


Fig. 31—Longitudinal Drum Water Tube Boiler

For plants requiring more than 200 h. p. of boilers there should be three or more units depending upon the maximum total load. The size and number should be so arranged as to provide one spare boiler at all times in case of repairs being required.

The question of the most economical working pressure for a boiler plant often arises and this can usually be determined from an examination of local conditions. If the plant is only for heating and no steam is required for cooking or manufacturing purposes, it is usually more satisfactory to operate the plant on low pressure, approximately 2 lb. per square inch. If steam is required for cooking at least 40 lb. per square inch should be used. In the latter case it is generally not advisable to split up the plant, operating one part on high pressure and one part on low pressure for heating, as too many complications will arise in design and operation. Besides, there is no particular economy gained by this arrangement. Steam for heating should be supplied through pressure reducing valves. The only difference from the standpoint of economy is that with the higher pressure there is a correspondingly higher temperature of steam and a slight increase in radiation losses from the exposed parts of the boiler.

Chimneys

To determine the size of a chimney for any plant, the height of the same above the boiler grates should be considered. When cast iron sectional boilers are used, Thompson gives the following formula for determining the grate area when the height is known.

$$A = \frac{G \times .75}{\sqrt{H}}$$

In which

A = Area of chimney in square feet.

G = Area of grate in square feet.

H = Height of chimney in feet.

Kent gives the following formula for determining the size of the stack in large plants when the boiler capacity is given in horse power.

$$H.\ P. = 3.33(A - .6\sqrt{A})\sqrt{H}$$

In which

H. P. = Horse power.

A = Area in square feet.

H = Height of stack in feet.

This is on a basis of five pounds of coal burned per square of grate per hour.

Table No. 17 gives the capacities of stacks in horse power for various heights and sizes determined from the above formula.

TABLE XVII

Stack Capacities in Horse Powers

Dia. in In.	Area in Sq. Feet	Height of Chimney in Square Feet 50	60	70	80	90	100	110	125	150	175	200	225	250	300'
18	1.77	23	25	27	29	...	..	....	....	....	....	....	....	....	....
21	2.41	35	38	41	44	...	...	....	....	....	....	....	....	....	....
24	3.14	49	54	58	62	66	...	....	....	....	....	....	....	....	....
27	3.98	65	72	78	83	88	...	....	....	....	....	....	....	....	....
30	4.91	84	92	100	107	113	119	....	....	....	....	....	....	....	....
33	5.94	..	115	125	133	141	149	156	....	....	....	....	....	....	....
36	7.07	..	141	152	163	173	182	191	204	....	....	....	....	....	....
39	8.30	..	...	183	196	208	219	229	245	268	....	....	....	....	....
42	9.62		...	216	231	245	258	271	289	316	342	....	....	....	....
48	12.57	..	...	...	311	330	348	365	389	426	460	492	....	....	....
54	15.90	..	...	...	...	427	449	472	503	551	595	636	675	....	....
60	19.64	..	...	...	...	536	565	593	632	692	748	800	848	894	....
66	23.76	..	...	...	...	...	694	728	776	849	918	981	1040	1097	1201
72	28.72	..	...	...	...	...	835	876	934	1023	1105	1181	1253	1320	1447
78	33.18	..	...	...	...	...	...	1038	1107	1212	1310	1400	1485	1565	1715
84	38.48	..	...	...	...	...	...	1214	1294	1418	1531	1637	1736	1830	2005

CHAPTER XV

HOT WATER SYSTEMS

ALL computations given in Chapter IV on the subjects of heat and heat losses from buildings apply to hot water heating as well as to steam. Estimating the size of mains, radiators, and radiating surfaces, however, requires different treatment as the temperatures, volumes, etc., dealt with are different. The maximum temperature usually estimated for hot water heating is 180°. Assuming an average temperature of 180° in the radiator and a room temperature of 70°, the B. t. u. transmitted per square foot can be determined in the same manner as with steam. The temperature difference between the radiator and the room is 180—70 = 110°.

Referring to Chapter VI, it will be noted that a radiator temperature of 220 deg. and a room temperature of 70 deg. or a difference of 150 deg. was designated for convenience sake as standard conditions. It will further be noted that, to find the rate of transmission of heat from a radiator under any other conditions or other difference in temperature than the above, the B. t. u. per degree difference was first found by dividing the total B. t. u. per square foot of surface by the difference in temperature. These values are given in table No. VI of Chapter VI. It was also stated that the values given in table No. VI or the transmission per degree difference per hour increases or decreases as the temperature difference varies above or below the assumed standard conditions of 150 deg. difference; this variation being 2% for each 10 degrees. With water temperature of 180 deg. in the radiator and a room temperature of 70 deg. the difference is 110 deg. or 40 deg. below the standard. 2% for each 10 deg. gives a decrease of 8% in the values given in table No. VI. Table No. XVII gives these values for the various types of radiators as given in table No. VI with the 8 per cent. decrease for hot water.

The following values are for transmission per degree difference with a difference of 110 deg. To get the total value these quantities must be multiplied by 110 degrees which are given in table No. XIX.

TABLE XVIII

B. t. u. per Square Foot per Degree Difference: Hot Water Temperature, 180°; Room Temperature, 70°

Type of Radiator	Height of Radiator 22 in.	26 in.	32 in.	38 in.
1 col.	1.75	1.71	1.68	1.66
2 col.	1.66	1.61	1.57	1.54
3 col.	1.56	1.52	1.47	1.42
4 col.	1.47	1.43	1.38	1.33
Window Radiator		1.70		
Wall Radiator (horizontal)		1.79		
Wall Radiator (vertical)		1.75		
Pipe coils		1.84		

TABLE XIX

B. t. u. Per Square Foot Hot Water. Water Temperature 180°; Room Temperature 70°

Type of Radiator	22 in.	26 in.	32 in.	38 in.
1 col.	193	188	185	183
2 col.	183	177	173	169
3 col.	172	167	162	156
4 col.	162	157	152	146
Window Radiator		187		
Wall Radiator (horizontal)		197		
Wall Radiator (vertical)		193		
Pipe coils		202		

After the total heat loss from the building has been determined as described in Chapter IV, the square feet of radiation is found by dividing this heat loss by value in the above table corresponding to the type and height of radiator selected.

Drop in Temperature

The force which produces the circulation in gravity hot water systems is the difference in weight between the water in the supply pipe and in the return pipe. This difference in weight is caused by the difference in temperature between these two columns of water. The water in the return pipe is colder, being cooled while passing through the radiator. The colder water therefore flows down the return riser to the boiler and is replaced by the warmer water in the supply pipe from the boiler; thus the circulation is continuous so long as a fire if kept in the boiler to reheat the returning water. The amount which the water is cooled or the drop in temperature depends upon the size of the flow and return mains. If the mains are large the velocity of water through the radiator will be more rapid than with small pipes and consequently the drop in temperature of the water in passing through will be less.

The drop in temperature is usually estimated at above 15 to 20 deg. for the entire system. With an assumed drop the quantity of water which must flow through a radiator can be determined. Assume a 2 column radiator 32 inches high. From table No. XIX the B. t. u. per square foot with the average water temperature of 180 deg. is found to be 173 B. t. u. If the temperature drop is 15 deg. then each pound of water flowing through the radiator will give up 15 B. t. u. As the transmission is 173 B. t. u. per sq. ft. per hour, then for each square foot of surface, there must be supplied $\frac{173}{15}=11.5$ pounds of water per hour or $\frac{11.5}{60.5}=.19$ cubic feet of water per hour. If the velocity of flow in the supply and return pipes to the radiators is known the size of the pipes can be determined. The velocity of flow depends upon the height of the radiator above the boiler and also the distance of the radiator from the boiler or the horizontal run. To illustrate; assume that 300 sq. ft. of radiation is to be supplied. If the velocity of flow in the supply pipe is .8 feet per second what would be the necessary size of pipe with 15 deg. drop? From the above the quantity of water per sq. ft. per hour is .19 cubic feet.

For 300 sq. ft. therefore, there will be $.19\times300=57$ cu. ft. per hour or $\frac{157.5}{3600}=.016$ cu. ft. per second.

Using the formula:

$$A=\frac{Q}{V}$$

$$A=\frac{.016}{8}=.02\text{ sq. ft.}$$

$$=.0219\times144=2.88\text{ sq. in.}$$

The area of a 2 in. pipe is 3.14 sq. in. therefore this would require a 2 in. pipe.

The pipe sizes for any system may be designed on a velocity basis by determining the height of the various radiators, length of runs, number of fittings, etc., and from this determining the velocity. This method requires considerable computation, however, and for small gravity systems it will be sufficiently accurate to size the mains from tables given later in the same manner as described for steam systems.

Piping Systems

There are various piping systems for hot water the same as in steam circulation which may be classified briefly as follows:

Two pipe, basement main systems, in which both the flow and return mains are in the basement supplying risers and radiators above.

Two pipe, down feed system, in which the flow mains are in the attic and the return mains are in the basement. With this system, one main supply riser extends from the boiler up to the attic where it distributes to the various down feed risers. The return risers start from the top floor radiators and extend down connecting with the return main in the basement. This system may be designed as a single pipe riser, the supply acting also as the return. The riser should then be the same size from top to bottom and of sufficient size to supply the entire amount of radiation on the riser.

One pipe circuit system which consists of one main starting out from the boiler and making a complete circuit in the basement, returning again to the boiler. This main is run full size for its entire length and all radiator and riser connections both supply and return are taken from it. The supply connections are taken from the top of the main and the return connections enter it at the side. Special fittings are often used in this system in taking the connections from the main to facilitate the flow.

The two pipe basement main system is generally used for residence work. The down feed system is preferable in many cases but usually there is not sufficient available space in the attic.

The following table given in N. J. Thompson's book on Mechanical Equipment of Federal Buildings may be used for residences and buildings of moderate size for a two pipe basement main system.

The first column gives the sizes of radiator tappings and connections.

The next four columns give the square feet of radiation to be supplied by the corresponding pipe sizes for first to fourth floor radiators.

TABLE XX

Pipe Size	1st Floor	2nd Floor	3rd Floor	4th Floor
¾	40	50	60	70
1	70	80	90	100
1¼	110	120	135	150
1½	180	195	210	230
2	300	350	400	500

In connection with table, No. XX, the following equalizing table, No. XXI, should be used to determine the size of risers and basement mains.

TABLE XXI

Size of Pipe	Equivalent carrying capacity	Size of Pipe	Equivalent carrying capacity
½ in.	2	3 in.	175
¾ in.	5	3½ in.	260
1 in.	10	4 in.	380
1¼ in.	20	5 in.	650
1½ in.	30	6 in.	1050
2 in.	60	7 in.	1600
2½ in.	110	8 in.	2250

In order to size the mains and risers the two tables should be used. The radiator tappings and connections for each radiator in the building are first sized from table No. XX. The equivalent area or carrying capacity of these tappings given in table No. XXI are then added together and the sizes of the mains and risers thus determined.

To illustrate, assume two risers supplying radiators on four floors as shown in Fig. 32. The amount of radiation in each radiator is given and the radiators are also indicated by letters.

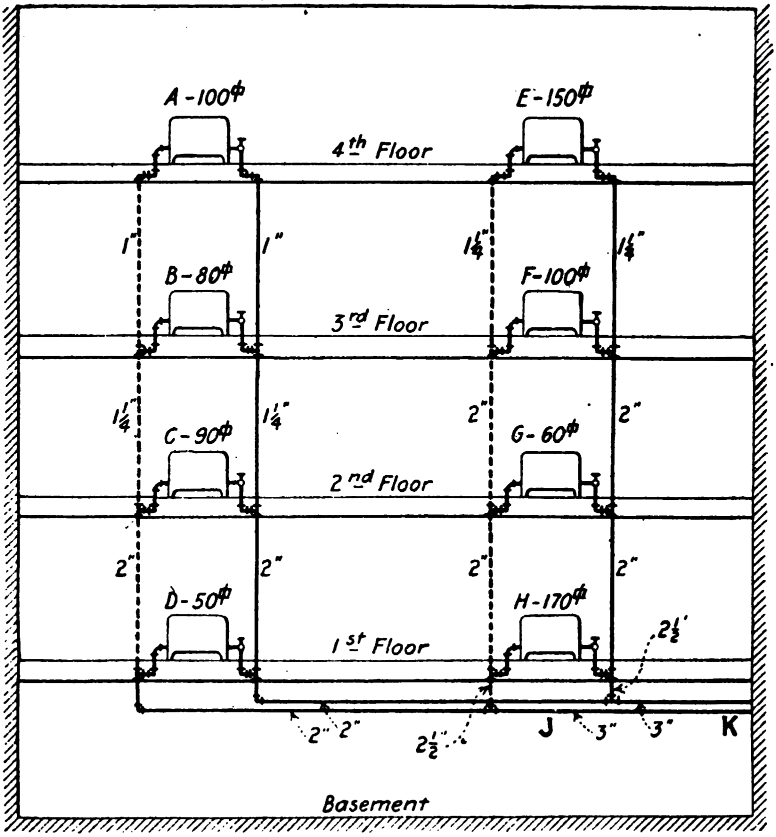


FIG. 32

Referring to table No. XX, the radiator connections should be as follows:

A = 1 in.
B = 1 in.
C = 1¼ in.
D = 1 in.
E = 1¼ in.
F = 1¼ in.
G = 1 in.
H = 1 ½ in.

These connections refer to the sizes of radiator tappings and connections, and size of riser where one radiator only is supplied.

To determine the size of the risers and mains refer to table No. XXI. The risers from radiator B to A will be the same as the radiator connection, or 1 in. Adding the equivalent areas of the tappings for the remainder of the riser we have:

A = 1 in. = 10
B = 1 in. = 10
20 = 1¼ in.
C = 1¼ in. = 20
40 = 2 in.
D = 1 in. = 10
50 = 2 in.

The riser between the second and third floors, or between radiators C and B, should be 1¼ in; between D and C, 2 in.; below D, or that part of the main between I and J, should be 2 in.

Proceeding with the other riser in the same manner, the connection between radiators F and E should be 1¼ in.

E = 1¼ in. = 20
F = 1¼ in. = 20
40 = 2 in.
G = 1 in. = 10
50 = 2 in.
H = 1½ in. = 30
80 = 2½ in.

The riser between G and F should be 2 in. Between H and G, 2 in. Between the main in the basement and radiator, H, should be 2½ in. To determine the size of the main beyond J or between J and K, the total sum of the equivalent areas for the first riser up to the point J is 50. At this point the equivalent area of the second riser which is 80, should be added, making a total of 130. This section should therefore be a 3 in. pipe. Proceed in this manner for the entire system back to the boiler. It must be remembered that the quantities given in table No. XXI do not represent square feet of radiation but are simply equivalent carrying capacities of these various pipes.

If the two pipe down feed system is used the size of radiator connections and tappings should be the same on all floors, that is, the tapping for any particular radiator should be the same, regardless of the floor on which it is located. To determine these sizes use table No. XX. The particular column of this table to use is the one corresponding to the height of the building in question. If the building is four stories high the radiator connections should be sized from the four story column, or a two story building should be sized from the two story column.

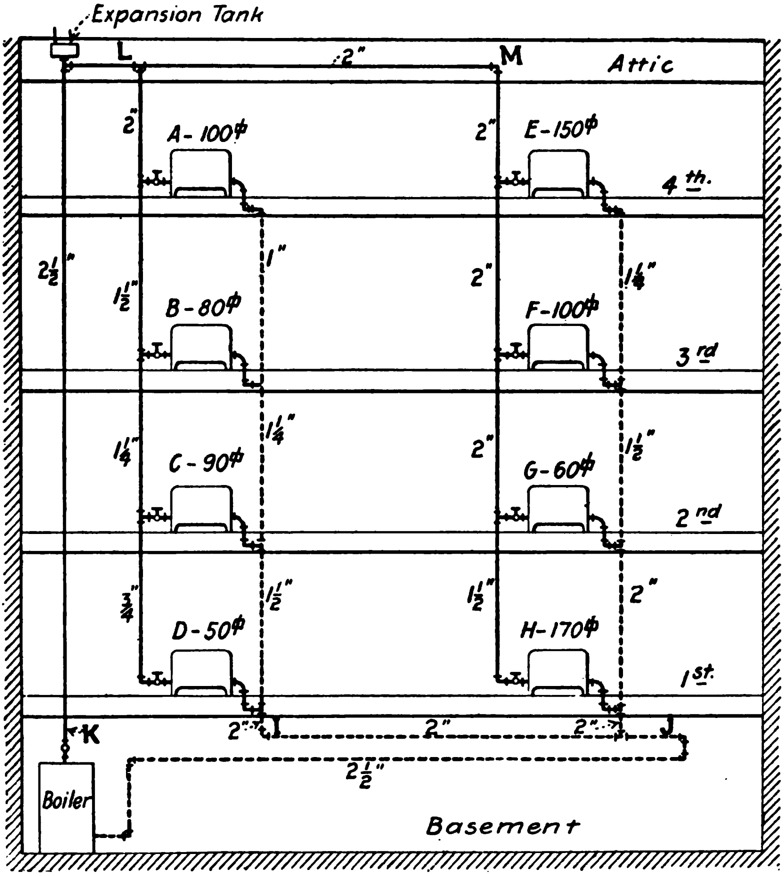


FIG 33

The reason for this may be more readily seen by referring to Fig. 33, which shows two down feed risers supplying radiators on each floor of a four story building. The size of the radiators have been assumed the same as in Fig. 76, as a means of comparison of pipe sizes for the two systems.

This being a four story building the last column in table No. XX should be used for all radiators. The tappings and radiator connections would, therefore, be as follows:

A = 1 in.
B = 1 in.
C = 1 in.
D = ¾ in.
E = 1¼ in.
F = 1 in.
G = ¾ in.
H = 1½ in.

The risers and mains should be sized in the same manner as before by the use of table No. XXI. The flow and return must be sized separately as they handle different amounts of radiation. To size the flow for the first riser, start at the bottom. The connection between radiators C and D should be ¾ in. Adding the areas as before:

D = ¾ in. = 5
C = 1 in. = 10
15 = 1¼ in.
B = 1 in. = 10
25 = 1½ in.
A = 1 in. = 10
35 = 2 in.

For the return riser start from the top:

A = 1 in. = 10.
B = 1 in. = 10
20 = 1¼ in.
C = 1 in. = 10
30 = 1½ in.
D = ¾ in. = 5
35 = 2 in.

For the flow of the second riser starting at the bottom:

H = 1½ in. = 30.
G = ¾ in. = 5
35 = 2 in.
F = 1 in. = 10
45 = 2 in.
E = 1¼ in. = 20
65 = 2½ in.

This last connection, from L to E, being only slightly over the carrying capacity of a 2 in. pipe, it would probably be safe to use a 2 in. pipe in this case.

For the return riser starting at the top:

$$\begin{array}{ll}
E = 1\frac{1}{4} \text{ in.} = & 20. \\
F = 1 \text{ in.} = & 10 \\
 & \overline{30} = 1\frac{1}{2} \text{ in.} \\
G = \frac{3}{4} \text{ in.} = & 5 \\
 & \overline{35} = 2 \text{ in.} \\
H = 1\frac{1}{2} \text{ in.} = & 30 \\
 & \overline{65} = 2\frac{1}{2} \text{ in.} \text{ — use 2 in.}
\end{array}$$

Adding the above sums to obtain the size of the flow main, we have $35+65=100=2\frac{1}{2}$ in. pipe. These sizes have been indicated on both sketches. It will be noted from examination of the two systems that the pipe sizes for the down feed system shown in Fig. 33 are slightly smaller than in Fig. 32.

It will further be noted from examination of Fig. 33 that the flow and return mains have been designed so as to obtain what may be called "parallel flow." Instead of starting the return main at the point J, running back to I and then to the boiler, the return starts at I, extends to J and then back to the boiler. The direction of flow in the section I-J is parallel to the flow in L-M. It can therefore be seen that the drop in pressure between the points L and I will be the same as the drop in pressure between M and J. The flow will therefore be more uniform with this arrangement. It usually requires a little more

TABLE XXII

Square Feet Radiation

Pipe Size	200 Foot Run	300 Foot Run
2	200	180
2½	400	300
3	600	500
3½	900	700
4	1200	1000
5	2300	1800
6	3600	3000
7	5200	4000
8	6800	6000

piping but this additional cost will be more than offset by the advantage gained.

For a one pipe circuit system, N. S. Thompson gives the preceding table, No. XXII, for determining the size of the main in the basement.

The total radiation should first be determined and the entire length of the circuit. With this data the size of pipe to use for the main can be determined from the above table. The size of radiators connections and risers should be determined in the same manner as described for the two-pipe basement main system by the use of tables No. XX and No. XXI.

Expansion Tanks

Expansion tanks are necessary on all hot water systems to allow for the expansion of the water when heated. These tanks simply act as reservoirs to take care of the additional volume of the water when heated to its maximum temperature. The tank should be located in the attic, several feet above the highest radiator in the building. A connection is taken usually from the highest point in the piping system and carried to the bottom of the tank. In the top of the tank, a vapor line or vent to the atomsphere is provided and also an overflow line to the sewer. To estimate the size of the expansion tank, the total quantity of water in the system must be determined. This is obtained by getting the volume of all the radiators, piping and boilers in the system. Then determine the increase in volume of this water for the entire range of temperature. Assuming the lowest temperature of water at 50° the weight per cubic foot at that temperature is 62.4 pounds. At 200 degrees temperature the weight per cubic foot is 60 pounds. The decrease in weight for this range is 2.4 pounds. The increase in volume is inversely proportional to the increase in weight therefore the percentage of increase in volume is $\frac{2.4}{60} = .04$ or 4 per cent. The total volume of water in the system multiplied by .04 will give the increase in volume in cubic feet. Table No. XXIII worked out on this basis gives the sizes of expansion tanks for various amounts of radiation.

With an open expansion tank the water in the system is under atmospheric pressure and therefore, will boil at a temperature of 212 degrees which is the maximum temperature that can be obtained under these conditions. There are a number of patented devices on the market for increasing this temperature.

TABLE XXIII

Sq. ft. of radiation	Size of Tank in inches
300	12 in. x 20 in.
500	12 " x 30 "
700	14 " x 30 "
950	16 " x 30 "
1300	16 " x 36 "
2000	16 " x 48 "
3000	18 " x 60 "
5000	20 " x 60 "
6000	22 " x 60 "

This is accomplished by installing some form of safety valve or mercury seal on the vent pipe from the expansion tank. This puts the system under as much pressure as the valve is set for as the water begins to expand and allows the temperature to be increased correspondingly. This is often times done in systems where the radiation is not sufficient to properly heat the building or the piping is too small to obtain proper circulation.

Forced Circulation of Hot Water

In the forced circulation system of hot water heating the circulation of water is produced by means of a power-driven pump. This pump is usually of the centrifugal type and may be driven by an electric motor, engine or turbine as the case may require. This system is often used on large buildings and may be used to advantage in buildings which are spread out over large areas. Much higher velocities and smaller pipes may be used with this system than with the gravity circulating system. The radiation is usually estimated on the same basis as for gravity circulation, that is, for a maximum temperature of 180 degrees and about 20 degrees drop in the radiator. The temperature drop in the radiator however, may be controlled from the power plant by controlling the velocity of flow.

This system may be used either with exhaust steam or with heating boilers. If exhaust steam is used the water is heated by means of a water heater of the closed type. This heater is usually supplemented with a secondary heater using live steam. If heating boilers, are used, the water is circulated directly through the boilers in the same manner as in the gravity system.

The piping system should be so designed as to obtain parallel flow in the supply and return mains if possible that is, to have the velocity of flow in the supply and return mains always in the same direction. This scheme can be carried out for an entire system both risers and mains in the down feet system which is recommended for forced circulation.

In designing the piping, the best method to follow is to have a uniform drop in pressure per unit length throughout the entire system. This will cause a lower velocity to be used in the smaller pipes than the larger pipes.

Table No. XXIV compiled by W. L. Durand is worked out on a uniform drop in pressure basis. The velocity is given in feet per second and the radiation supplied in square feet with 20 degrees drop.

The total drop in pressure for the system should first be selected and from the total length of run, the drop per 100 feet may be determined. The column corresponding to this drop should then be used for determining the pipe sizes. The drop in pressure in the above tables is given in feet.

To determine the horse power necessary to drive the pump the total quantity of water must be determined. The head through which this must be lifted is the head equivalent to the drop in pressure. With these two factors the actual horse power may be determined.

TABLE XXIV

Velocity and Volume of Water and Radiation for Different Friction Heads

Size Pipe in.		Friction Head per 100 Ft. Length of Pipe in Feet									
		1	1¼	1½	2	2½	3	3½	4	4½	5
1	Vel.	1.21	1.36	1.50	1.75	1.96	2.16	2.34	2.51	2.68	2.83
	Cu.ft.	0.436	0.490	0.540	0.630	0.695	0.777	0.841	0.904	0.965	1.02
	Rad.	186	210	230	270	297	332	360	386	412	435
1¼	Vel.	1.50	1.68	1.85	2.14	2.41	2.65	2.88	3.09	3.29	3.47
	Cu.ft.	0.936	1.05	1.15	1.33	1.50	1.65	1.80	1.93	2.05	2.16
	Rad.	400	450	490	570	640	705	770	825	875	920
1½	Vel.	1.60	1.80	2.00	2.31	2.61	2.88	3.11	3.34	3.58	3.76
	Cu.ft.	1.35	1.52	1.69	1.95	2.21	2.44	2.63	2.82	3.03	3.18
	Rad.	575	650	720	830	945	1040	1125	1200	1300	1360
2	Vel.	1.90	2.13	2.35	2.73	3.08	3.40	3.69	3.93	4.20	4.44
	Cu.ft.	2.81	3.15	3.48	4.04	4.56	5.03	5.46	5.82	6.22	6.57
	Rad.	1200	1350	1490	1720	1950	2150	2330	2480	2660	2810
2½	Vel.	2.14	2.40	2.64	3.08	3.48	3.81	4.15	4.45	4.73	5.00
	Cu.ft.	4.28	4.80	5.28	6.16	6.96	7.62	8.30	8.90	9.46	10.00
	Rad.	1830	2050	2260	2630	2980	3260	3540	3800	4950	4270
3	Vel.	2.44	2.75	3.02	3.53	3.98	4.40	4.76	5.10	5.40	5.72
	Cu.ft.	7.51	8.47	9.30	10.9	12.3	13.5	14.7	15.7	16.6	17.6
	Rad.	3210	3620	3980	4650	5250	5760	6270	6700	7100	7500
3½	Vel.	2.67	3.00	3.31	3.86	4.33	4.80	5.20	5.56	5.90	6.25
	Cu.ft.	11.0	12.4	13.6	15.9	17.8	19.8	21.4	23.0	24.4	25.8
	Rad.	4700	5300	5800	6800	7600	8450	9150	9800	10,400	11,000
4	Vel.	2.89	3.25	3.58	4.18	4.70	5.15	5.60	6.00	6.40	6.75
	Cu.ft.	15.3	17.2	19.0	22.2	24.9	27.3	29.7	31.8	33.9	35.8
	Rad.	6530	7350	8100	9500	10,600	11,650	12,700	13,600	14,500	15,600
4½	Vel.	3.08	3.47	3.80	4.42	5.00	5.50	5.96	6.40	6.80	7.20
	Cu.ft.	20.4	23.1	25.2	29.4	33.2	36.6	39.6	42.5	45.2	48.0
	Rad.	8700	9900	10.800	12,600	14,200	15,600	16,900	18,150	19,300	20,500
5	Vel.	3.30	3.72	4.10	4.80	5.40	5.93	6.45	6.90	7.35	7.75
	Cu.ft.	27.5	31.0	34.2	40.0	45.0	49.4	53.7	57.5	61.1	64.5
	Rad.	11,750	13,250	14,600	17,100	19,200	21,100	23,000	24,600	26,100	27,500
6	Vel.	3.75	4.22	4.67	5.40	6.10	6.70	7.25	7.80	8.30	8.75
	Cu.ft.	45.1	50.8	56.2	65.0	73.5	80.6	87.3	94.0	100	105
	Rad.	19,300	21,700	24,000	27,800	31,400	34,500	37,200	40,100	42,700	45,000
7	Vel.	4.10	4.61	5.08	5.90	6.65	7.30	7.90	8.45	9.00	9.50
	Cu.ft.	66.2	74.5	82.0	95.3	107	118	127	136	145	153
	Rad	28,200	31,800	35,000	40,700	45.750	50,500	54,300	58,200	62,000	65,500
8	Vel.	4.38	4.92	5.43	6.35	7.15	7.85	8.50	9.15	9.75	10.4
	Cu.ft.	91.5	103	113	133	149	164	177	191	204	217
	Rad.	39,000	44,000	48,300	57,000	63,700	70,000	75,700	81,600	87,100	93,000
10	Vel.	5.02	5.68	6.25	7.25	8.20	9.00	9.75	10.5	11.1	11.8
	Cu.ft.	165	187	206	238	270	296	320	345	365	388
	Rad.	70,400	80,000	88,000	101,500	115,200	126,000	137,000	147,000	156,000	166,000

CHAPTER XVI

FURNACE HEATING

DURING the past few years, furnaces and furnace heating for residences have been rapidly replaced by steam and hot water heating, though in some localities this system is still quite extensively used, particularly in small and inexpensive residences. If the system is well designed and installed with a good type of furnace, very satisfactory results should be obtained.

In designing a hot air system of heating, the method of procedure should be the same as for either steam or hot water. The heat losses for the various rooms should first be determined as described in Chapter IV, making all necessary allowances for exposure, etc. After these losses have been determined, the next step is to estimate the quantity of air that must be delivered to the room to supply the necessary heat. With a temperature of zero degrees outside the air entering the room is usually estimated at a temperature of 120 degrees. This air, cooling from 120 degrees to the temperature of the room, 70 degrees, must give up the quantity of heat necessary to counteract the heat loss through walls and windows and by leakage as the quantity of heat given up by this air must equal the total heat losses from the room.

For indirect heating systems as explained in Chapter XIII the air is assumed to enter the room at 120 degrees and gives up the necessary heat by cooling to 70 degrees. Therefore formula (5) given in that chapter for determining the quantity of air which must enter the room will also apply for determining the quantity of air for a hot air system. This formula was as follows:

$$Q=\frac{H\times 55}{50}$$

in which Q equals the quantity of air per hour and H is the total B. t. u. loss from the room per hour. Applying this formula, therefore gives the total quantity of air per hour and dividing by 60 gives the cubic feet of air per minute.

To determine the size of pipes leading to the various rooms a velocity of from 200 to 250 feet per minute is assumed for the pipes to the first floor depending on the length of horizontal run, and a velocity of 300 to 350 feet per minute in pipes to the second floor rooms. Applying the formula $Q = A\ V$, the size of the pipes can then be determined. The net free area through the registers should be about 25% more than the area of the duct.

To determine the size of the furnace the efficiency of the same must be considered in the same manner as for boiler. For ordinary practice an efficiency of about 50% to 60% should be assumed.

The average coal used in furnaces will give about 13000 to 14000 B. t. u., therefore, the actual quantity of heat utilized per pound of coal will be about 7000 B. t. u. per pound. If the total heat loss from the building to be heated is divided by 7000 the result will be the number of pounds of coal that must be burned per hour.

Grate Surface

For the average furnace the rate of combustion should be assumed at about 4 to 5 pounds per square foot of grate per hour, depending on the size of the furnace. Knowing the total quantity of coal that must be burned the size of the grate can therefore be determined.

Heating Surface

The heating surface is that part of the furnace that is in contact with the hot gases or fire on one side and the outside air being warmed and delivered to the room on the other side. The amount of heating surface in a furnace should have a definite relation to the grate surface. The ratio of the square feet of grate area to the square feet of heating surface is usually taken at about 1 to 20.

Cold Air Duct

The area of the cold air duct leading to the furnace should be about 80 to 90 per cent of the total combined area of the warm air ducts. It is good practice to have two cold air ducts leading from each side of the house in order to eliminate as far as possible the tendency of the winds from interfering with the air circulation. If this cannot be done the cold air inlet should be placed on the north or west side, so that the prevailing winds will assist in the circulation of air through the furnace.

A recirculating duct should also be provided so that at times the cold air may be closed and the air be drawn from within the building.

The furnace should be located low enough so that the hot air pipes may have a good pitch upward in the direction of the air travel.

UNIV. OF CALIFORNIA

CHAPTER XVII

HOT-BLAST HEATING

THE term mechanical ventilating system is applied only to such cases where the air is moved and distributed by mechanical means, with no reference to the force of gravity. In present practice where large volumes of air are to be handled no consideration is given to gravity in the design.

In taking up the question of ventilation a distinction should first be made between the ventilating system proper and the combination of a heating and ventilating system or where heating may be the primary object.

In the first case the entire equipment for heating, moving and distributing the air is called the ventilating system. In conjunction with this there may or may not be, as the case may require, other means of heating in the form of direct radiation. In the latter case, however, where the primary object is to provide heat and the ventilation may be a secondary consideration, the entire equipment used to heat, move and distribute the air is properly termed the hot-blast system of heating.

The Hot-Blast System

The hot-blast system is oftentimes used in large factories, foundries, car shops, paper mills, etc., where the building comprises one room of large cubical contents. Fig. 34 shows a typical arrangement of this system. The air is drawn in by the fan over heating coils which heat it to the proper temperature. It is then forced through the distributing ducts to the various parts of the building.

One of the advantages of this system is that the room can be brought up to temperature much more quickly than with direct radiation. It also has the advantage, in addition to providing ventilation when required, of producing a more uniform temperature throughout the room than can be produced with direct radiation. If the building has high ceilings, as is usually the case with car shops, foundries, etc., when heated entirely with direct radiation, the upper portion of the room will be at a very high temperature when comfortable working conditions are produced near the floor level. The result of this is that at the upper portion of the room there will be a much more rapid loss of heat through the exposed walls and roof than with a normal temperature and consequently more steam is required to maintain the desired temperature.

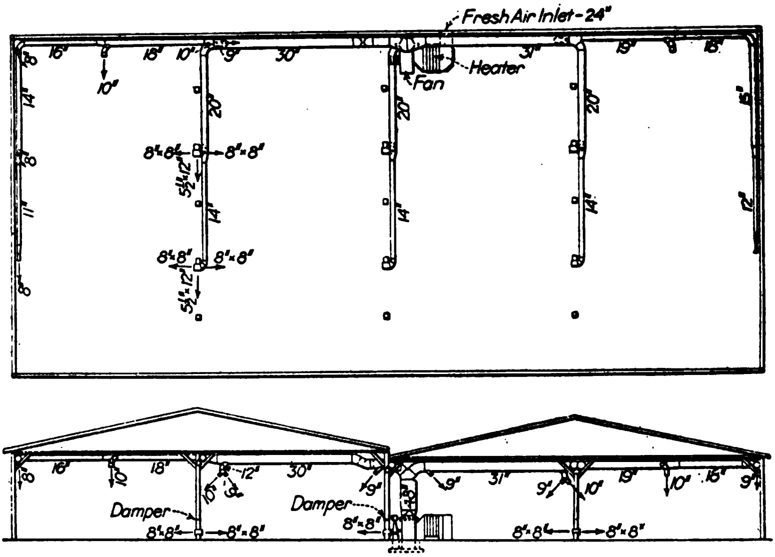


FIG. 34 – TYPICAL FAN-BLAST INSTALLATION FOR A FACTORY

With the hot-blast system the ducts should discharge downward within a few feet of the floor. This brings the warm air down near the floor where it is needed. The outlets of the ducts should also be located from 15 to 25 ft. from the outside walls if possible. With this system there is no direct radiation installed, the entire heating being accomplished by means of the heated air.

Let it be assumed that the room under discussion is to be maintained at a temperature of 70 deg. during zero weather. A definite quantity of air is being blown into the room by means of the fan and therefore the same quantity of air must continually pass out of the room if the fan draws its supply from outside. The air leaving the room will be at the same temperature as the room, or 70 deg. in this case, and the heat contained in this air or heat added to it to raise it from 0 deg. to 70 deg. is lost as far as actual heating effect is concerned. If there were no other losses of heat from the room except that represented by the air passing out of the room it would then only be necessary to raise the temperature of the entering air to 70 deg. This condition would only be met by a room surrounded on all sides by rooms maintained at a temperature of 70 deg.

This air, however, coming in contact with the cold exposed walls and glass surface of the room, is cooled. It is therefore necessary to introduce the air at a temperature sufficiently high so that in cooling from this temperature to the required temperature it will give up just enough heat to counteract the cooling effect of the walls. In other words the heat given up by the air in cooling from its entering temperature to the room temperature of 70 deg. must equal the heat lost through all exposed walls and glass surface by conduction and radiation from an inside temperature of 70 deg. to an outside temperature of 0 deg.

Referring to the formula given in Chapter IV for determining the heat loss from buildings, we find that this formula is made up of three items, namely: heat loss from exposed walls; heat loss from exposed glass; heat loss through air change.

Applying this to the present discussion it can be seen that the third item, or the heat loss through air change, is represented by the air leaving the room. The amount of air leaving must be the same as the amount being blown in, and at a temperature equal to the room temperature. Therefore the heat supplied by the air above 70 deg. must equal the heat lost through walls and windows (not including air change). It must be remembered also that any other sources of loss, such as through floors, ceilings, etc., and allowances for exposure should be added to this. It is also advisable to add from 5 to 10 per cent for wind leakage, for on windy days, even with the air being blown in under pressure, there is apt to be a small amount of leakage of cold air into the room around windows. It is not necessary to make any allowance for high ceilings, for there will not be the excessive increase in temperature at the ceiling line with this system as with direct radiation.

As has been demonstrated in Chapter IV, the heat required to raise 1 cu. ft. of air 1 deg. is 1/55 B.t.u. Therefore, the heat given up by 1 cu. ft. of air cooling 1 deg. is 1/55 B.t.u. By cooling from an entering temperature of T_1 deg. to T deg. the B.t.u. given up will be $(T_1—T)\times 1/55$ or $\frac{T_1—T.}{55}$ If there are Q cu. ft. per hour entering the room the total B.t.u. given up will be

$$Q\times\frac{T_1—T}{55} \text{ or } \frac{Q\ (T_1—T).}{55}$$

Equating this to the heat loss from exposed surfaces (H) as demonstrated in the foregoing we have

$$\frac{Q\ (T_1—T)}{55}=H. \qquad (1)$$

which is the same as formula (3) given under indirect systems. Transposing and solving this for Q we have

$$Q=\frac{H\times 55}{T_1—T} \qquad (2)$$

which expresses the cubic feet of air per hour in terms of temperature drop and heat losses. By another transformation we have

$$T_1—T=\frac{H\times 55}{Q} \qquad (3)$$

which expresses the necessary temperature drop in terms of the cubic feet of air per hour and the heat losses per hour. Whether formula (2) or (3) is to be used in designing a system will depend upon which of the two unknown quantities, the cubic feet of air per hour or the temperature drop, is to be assumed.

The first step in the solution of the problem is obviously the determination of (H), which should be solved as explained under indirect systems by the formula

$$H=(GK_g+WK_w)\ T$$

including as pointed out in the foregoing losses through floors, ceilings, exposures, etc.

The next step is the determination of one of the two unknown quantities Q, quantity of air or $(T_1—T)$ in order to solve for the other. The quantity of air is usually assumed from the character of the building under consideration, and it is customary to designate this as a certain number of air changes per hour.

The term air change means the number of times that the entire volume of air in the room will be changed every hour. Assume that a room has a cubical contents of 120,000 cu. ft. If 120,000 cu. ft. of air are blown into the room per hour or 2000 cu. ft. per minute the room is said to have one air change per hour. Or expressing this in minutes, the volume of the room is changed every 60 minutes, and is therefore, stated to have a 60-min. air change. If 360,000 cu. ft. per hour or 6000 cu. ft. per minute are admitted the volume is changed three times per hour and therefore has three air changes per hour. The volume is changed every 20-min. and may be stated as a 20-min. air change.

For the types of buildings now under discussion in no case should the air be changed less than once every hour and in general from two to four changes per hour should be used, depending on how much ventilation is necessary. The smaller the number of air changes that can be safely used, however, the less will be the first cost of installation and also the cost of operation will be decreased.

In machine shops, car shops, etc., where the cubical contents is usually large compared with the number of occupants and where there are no objectional gases or odors to be removed the number of air changes may be small and in some cases one change per hour may be used with safety. It may be found that with the assumed air change the temperature of the entering air is too high, particularly in buildings with a large amount of glass and exposed wall surface. If the temperature is above 140 deg. the amount of air should be increased, or in such a case the temperature could be fixed at 140 deg. and inserting this in formula (2) the quantity of air to use can be determined.

In foundries, paper mills, etc., where there are objectionable gases to be removed, from three to four air changes per hour should be used, and in some cases considerably more than this may be necessary. The number of air changes to use having been determined and the volume of the room known, the cubic feet of air can be determined. Let N equal the number of air changes per hour, and V equal the volume of the room in cubic feet. Then $N \times V = Q$ cu. ft. per hour. Inserting this value of Q in formula (3) gives the value of $(T_1 - T)$. This quantity added to the temperature of the room gives the value of T_1 the temperature of the entering air. The question of determining the temperature to which air will be heated in passing through heating stacks under different conditions will be discussed in the succeeding chapters.

Problem

Assume a factory building, as shown in Fig. 35. The side and end walls are constructed of 8-in. concrete. The roof is constructed of slate on sheathing. The building is say, a machine shop and is to be heated to 60 deg. in zero weather. Determine the temperature at which the air must be introduced with 1½ air changes per hour.

Solution

Total exposed wall and glass surface on one side is

$$
\begin{aligned}
200 \times 15 &= 3000 \text{ sq. ft.} \\
200 \times 10 &= 2000 \text{ sq. ft.} \\
\hline
\text{Total} &= 5000 \text{ sq. ft.}
\end{aligned}
$$

On two sides: 5000 × 2 equals 10,000 sq. ft. Total exposed glass on one side,

$$
\begin{aligned}
180 \times 10 &= 1800 \text{ sq. ft.} \\
180 \times 6 &= 1080 \text{ sq. ft.} \\
\hline
\text{Total} &= 2880 \text{ sq. ft.}
\end{aligned}
$$

On two sides: 2880 × 2 equals 5760 sq. ft. Net wall on sides equals 10,000 minus 5760, equal 4240 sq. ft. Exposed wall on one end equals

$$
\begin{aligned}
15 \times 80 &= 1200 \text{ sq. ft.} \\
15 \times 60 &= 900 \text{ sq. ft.} \\
10 \times 40 &= 400 \text{ sq. ft.} \\
15 \times 20 &= 300 \text{ sq. ft.} \\
\hline
\text{Total} &= 2800 \text{ sq. ft.}
\end{aligned}
$$

On two ends equals 2800 × 2 equals 5600 sq. ft. Total wall 5600, plus 4240 equals 9840 sq. ft.

To find the roof surface the length of the sloping portion is found to be 25 ft. where the rise is 15 ft. and the projected length is 20 ft. The area of one section of the roof is therefore

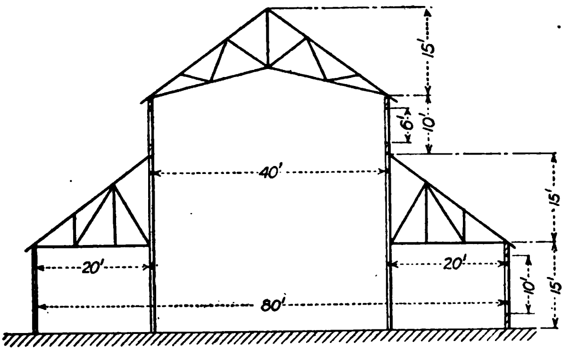


FIG. 35A—END VIEW

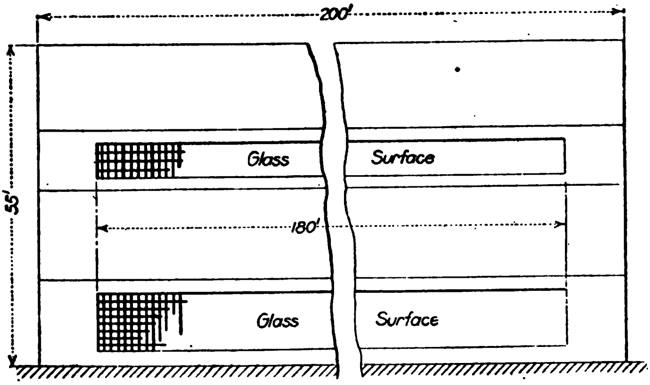


FIG. 35B—SIDE VIEW

DIAGRAMMATIC EXPLANATION OF HOW AIR CHANGES IN FACTORY ARE COMPUTED

25 × 200, equals 5000 sq. ft. For the four sections 5000 × 4 equals 20,000 sq. ft.

Referring to the table of wall factors given in Chapter IV the heat loss through 8-in. concrete is found to be 0.48 per degree difference. The heat loss through the walls is then 9840 × 0.48 equals 4723 B.t.u. per degree difference.

The factor for slate roof with sheathing is 0.38. The heat loss through the roof is 20,000 × 0.38 equals 7600 B.t.u. per degree difference.

The heat loss through the glass surface is 5760 B.t.u. per degree difference.

The total heat loss per degree difference is 4723 plus 7600 plus 5760, equals 18,083 B.t.u. The room is to be heated to 60 deg.; therefore 18,083 × 60 equals 1,085,000 B.t.u. loss from the room per hour.

To find the volume of the room multiply the area of the end by the length of the room. The area as given in the foregoing is 2800 sq. ft. and 2800 × 200 equals 560,000 cu. ft. With 1½ changes per hour 560,000 × 1½ equals 840,000 cu. ft. as the quantity of air to be delivered into the room by the fan.

Applying formula (3)

$$T_1 - T = \frac{H \times 55}{Q}$$

$H = 1{,}085{,}000$ B.t.u.

$Q = 840{,}000$ cu. ft.

$T = 60$ deg.

$$T_1 = 60 = \frac{1{,}085{,}000 \times 55}{840{,}000} = 71$$

$$T_1 - 60 = 71.$$

$T_1 - 71 + 60 = 131$ deg., the temperature at which the air must be admitted to maintain a temperature of 60 deg. in zero weather.

CHAPTER XVIII

DUCT SYSTEMS FOR FACTORY VENTILATION

AFTER the amount of air for heating and ventilating purposes has been determined, the duct system for distributing this air to the required points must be designed. The term "duct" as applied to ventilation refers only to the horizontal portion of the system. Any vertical riser carrying air either up or down is called a flue. Care should be taken to distinguish between ducts and flues, as complications may arise if this is not done.

To determine the size of ducts and flues, formula No. 3, derived in Chapter 8 must be referred to, which was as follows:

$$A = \frac{Q}{V}$$

In this formula A = area, Q = quantity, and V = velocity. In this formula only one of these quantities is at present known from assumptions already made—namely, (Q) the quantity of air. In order, therefore, to determine the area of any particular duct the velocity must be assumed. (The basis upon which these velocities are assumed will first be discussed.) The same fundamental principles which were explained in the flow of steam in pipes apply also to the flow of air in pipes and the determining factor is the same—namely, the drop in pressure caused by the friction of the air moving in contact with the surface of the pipe or duct. The velocities and pressures used in the distribution of air for ventilating purposes, however, are much less than are used in the distribution of steam.

There are several different methods of determining the velocity mathematically for the different conditions, and these will be discussed in succeeding chapters. The method usually adapted is to assume the velocities at the various points and those assumptions are based on what has been found satisfactory from actual experience.

The only advantage gained in using comparatively high velocities is that the ducts will be smaller, and consequently cost less and require less room. On the other hand, these higher velocities will produce greater friction and more pressure will be required to move the air. Therefore, what is gained in the first cost of the duct work will be partly lost by the higher first cost of the driving motor or engine. The cost of operation is also increased. It is, therefore, advantageous where possible to use comparatively low velocities.

The main duct is sometimes made the same size as the outlet of the fan and is decreased in size as the branches are taken off. This method will be found to give rather high velocities. The best results will be obtained if the velocity in the main duct does not exceed 1500 ft. per minute. For factory work this figure can be safely adopted as a standard for the first part of the system from the outlet of the fan to the first branch duct. At this point the velocity in the main duct should be decreased 50 to 75 ft. per minute depending upon the distance between this and the succeeding outlet. The velocity should be decreased in this manner and at about this same proportion at each outlet from the main duct. This will gradually decrease the loss in friction and at the same time increase the static pressure and thus produce the same flow of air from the outlets at the extreme ends of the duct as from those nearer the fan.

The velocity in the branch ducts should be decreased as soon as they leave the main duct. The velocity at the outlet into the room should not exceed 600 ft. per minute, and if in any case the air blows directly upon the occupants of the room, the velocity should not be more than 400 ft. per minute. In all duct work deflectors with indicating quadrants should be installed at all division points and butterfly or volume dampers should be installed near the outlet of each branch.

Problem

Design the duct system for a building 225 ft. long by 100 ft. wide, with a 30 ft. ceiling. The building to have four air changes per hour. The fan to be located at one end of the building and one main rectangular duct to be carried down the center of the building, and branches running out 25 ft. each way and discharge near the floor.

Solution

Fig. 36 shows a plan view of the building and a general arrangement of the system.

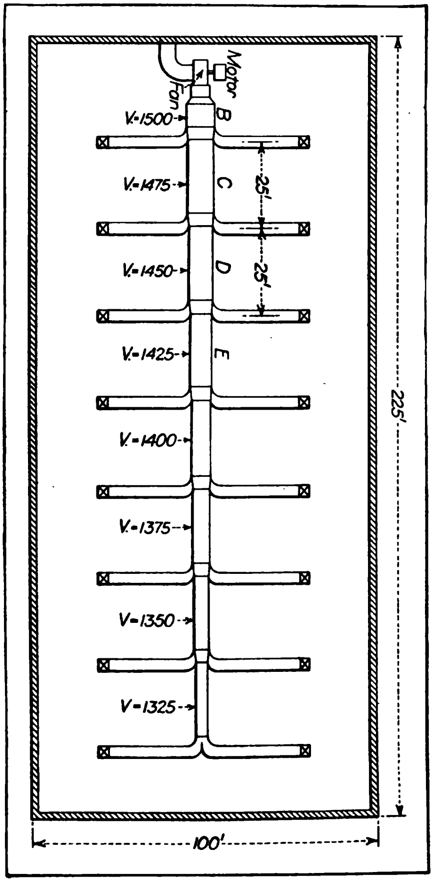


FIG. 36—DIAGRAMMATIC EXPLANATION OF METHODS OF PROPORTIONING MAIN DUCT IN FAIR BLAST HEATING SYSTEM

Volume of building $=225\times100\times20=450{,}000$ cu. ft. Four air changes per hour:

$450{,}000 \times 4 = 1{,}800{,}000$ cu. ft. per hour.

$1{,}800{,}000 \div 60 = 30{,}000$ cu. ft. per minute.

The building is 225 ft. long and with two outlets for every 25 ft. gives 16 outlets.

$30{,}000 \div 16 = 1875$ cu. ft. per minute per outlet.

The velocities to use at each section are indicated on the plan.

For section "B", $Q = 30{,}000$, $V = 1500$, $\therefore A = \dfrac{30{,}000}{1500} = 20$ sq. ft. $20 \times 144 = 2880$ sq. in.

Let the duct be 36 in. deep at this point, then $2880 \div 36 = 80$.

Section "B" should, therefore, be 80 in. wide by 36 in. deep.

Section "C"—$1875 \times 2 = 3750$ cu. ft. are discharged from the first two outlets.

Therefore, Q for section "C" equals $30{,}000 - 3750 = 26{,}250$ cu. ft. per minute.

$V = 1475$ ft. per minute.

$\therefore A = \dfrac{26250}{1475} = 17.8$ sq. in. area.

$17.8 \times 144 = 2563$ sq. in. area.

$2563 \div 36 = 71$ in.

Section "C" should, therefore, be 36 in. × 71 in.

Assume a velocity of 1000 ft. per minute in the horizontal branch ducts.

$Q = 1875$. $V = 1000$.

$\therefore A = \dfrac{1875}{1000} = 1.88$ sq. ft.

$1.88 \times 144 = 271$ sq. in.

Make the first duct 10 in. × 28 in. In the vertical flue leading down to the floor decrease the velocity to 500 ft. per minute. This velocity is one-half the velocity in the horizontal portion, therefore, the area will be twice this area of the horizontal duct. Make this duct 20 in. × 28 in.

Problem

Determine the areas of the remaining sections of the system. Would the duct work cost less if there were two main distributing ducts on each side of the building directly over the vertical flues?

CHAPTER XIX

SCHOOL VENTILATION

THE problem of school ventilation must be handled in an entirely different manner from the hot-blast system described in the previous chapter. The quantity of air is determined entirely from the pupils in each classroom. Certain experiments have been made as to the quantity of air necessary for breathing and the amount of impurities given off, which consists principally of carbon dioxide. From these experiments the quantity of air per pupil has been arbitrarily fixed at 30 cu. ft. per minute. Each classroom has a given seating capacity and the quantity of air per minute for each class room can be determined by multiplying the number of pupils by 30. The room temperature is usually fixed at 70 deg.

There are two distinct methods used for heating and ventilating; first, where the air is admitted into the room at a temperature sufficiently above 70 deg. to provide the necessary heat, and second, where the air for ventilation is admitted into the room at 70 deg. and the heating is accomplished by direct radiation placed under the windows in the usual manner. This second method is the most satisfactory, as it eliminates the necessity of blowing the air in at a high temperature. Furthermore, it is only necessary to operate the fan during school hours and still provide heat in the classrooms before and after sessions, which is usually desirable.

For the first method, the temperature of the entering air is estimated in the same manner as given in the hot-blast system. With the latter method, the air entering at 70 deg., leaves the room at 70 deg. and therefore, neither gives up heat to the room nor takes away any heat. It can be seen, however, that in estimating the amount of direct radiation for the room, the question of air change need not be considered as the air entering the room is preheated to 70 deg. by the heating stack of the ventilating system, whereas, in estimating for direct radiation where there is no ventilating system, the air must be considered as entering from the outside and a temperature of zero.

The amount of direct radiation necessary for this second method system is also not affected by the quantity of air admitted to the room, because as stated in the foregoing no heat is added or taken away by the air. If the amount of air is increased or decreased, the temperature of the room will not be affected, providing the temperature of the incoming air is maintained at 70 deg.

The formula for determining the amount of direct radiation is the same as given in a preceding chapter, namely $H = (G K_g + W K_w) T$ where H is the B.t.u. loss per hour and $T = 70$ deg. After H has been determined for the given condition its value is divided by the B.t.u. given off from the radiating surface per square foot and the result thus obtained is the required amount. Losses through any exposed floors or ceilings should be included in the determination and proper allowances made for north and west exposures and leakage.

Air Velocity

One of the most important things to guard against in the design of school ventilating systems is noise, and as a rule, it is one of the most difficult things to eliminate. It is very important that the velocity of air in all parts of the direct system be kept low and at no point should it exceed 1200 ft. per minute. In the main distributing duct from the outlet of the fan the velocity should be about 1000 ft. per minute. If the main duct is of any considerable length this velocity should be decreased toward the end as the branch ducts are taken off. This decrease should be from 25 ft. to 50 ft. per minute for each branch taken off, depending upon the distance between these branches.

After leaving the main duct, the branch ducts should be immediately increased in size so that the velocity is decreased to about 400 or 500 ft. per minute. As these branch ducts approach the register faces, they should again be gradually increased to the size of the register face. Sudden increases in size, partucilarly near the register face, should be avoided, as this is likely to produce an uneven velocity of air leaving the register.

The size of the register face should be such as to produce a velocity of not over 250 ft. per minute leaving the register. The State Board of New Jersey requires that the velocity at the register face shall not exceed 300 ft. per minute. If the velocity is estimated at 250 ft. per minute, this will allow a slight variation for unequal velocities at various parts of the register and still give sufficient area for the required amount of air. It is necessary in some cases where the flue or duct turns sharply at right angles to the register face, to put in deflectors at the bend in order to produce a uniform velocity at the register. Without deflectors, the air would take a course such as represented by the arrows in Fig. 37.

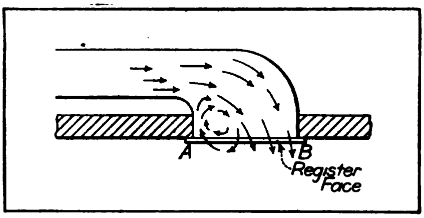


FIG. 37·—SHOWING EFFECT OF CHANGE OF DIRECTION ON AIR DELIVERY AT REGISTER

The air at the side B will leave the register face at a high velocity, due to the momentum of the air in the duct. It will often be found that at the side A the air will be flowing back through the register, showing that there are eddy currents formed as indicated by the arrows.

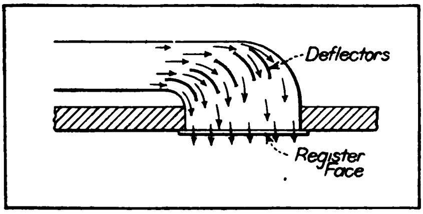


FIG. 38.—SHOWING HOW DELIVERY IS EQUALIZED WHEN DEFLECTORS ARE USED

The best results can be obtained by short, curved deflectors as shown in Fig. 38. These deflectors can be set in place temporarily when the duct work is constructed and with a little adjustment, after the system has been started up, a very uniform velocity can be obtained. The deflectors should not extend out to the register face as this space between the deflectors and the register provides an opportunity for the air to diffuse properly.

The supply register should be located not less than 8 ft.

from the floor and on an inside wall. Each classroom must be provided with an exhaust register which should be located near the floor. Both registers, if possible, should be on the same wall and at opposite ends of the room. This arrangement provides a thorough air removal from all parts of the room.

The exhaust register is usually made the same size as the supply and the flue should be of such size that the velocity of air is not more than 400 ft. per minute, estimating the same quantity of air leaving the room through the exhaust flue that enters through the supply.

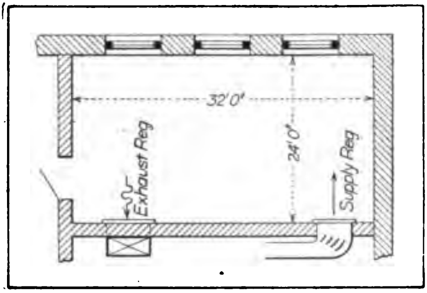


Fig. 39.—Plan of Classroom for Forty-six Pupils

No exhausting equipment is necessary for schools of ordinary size, such as an exhaust fan or aspirating coils. The vent flues are usually combined into one or more exhaust flues as their location may require and this flue discharges through a ventilator on the roof.

All toilet rooms should have individual exhaust flues and some form of exhausting equipment such as a small steam coil placed in the base of the flue or a small motor driven fan. No fresh-air duct from the ventilating system should be provided. This applies not only to school toilet rooms but also to toilet rooms in any class of buildings, as a plenum condition might be produced at times in the toilet rooms by this arrangement and the air forced out into other parts of the building.

Direct Radiation

For the direct radiation in the classrooms, wall coils constructed of wall radiation are very satisfactory and can be located conveniently under the windows. This arrangement not only takes up a small amount of space but also the coils can be so arranged as to cross an entire exposed wall and thus distribute the heating surface where it is most needed. This will produce a more uniform room temperature than if one large radiator is placed at one point. It is good practice to divide the coil into two units so that in mild weather only one unit need be turned on and thus give better regulation.

The arrangement of the heating stacks, blowers, etc., will be discussed in succeeding chapters.

Problem

Give the size of wall coil composed of two units, the size of registers and supply duct for a forty-six pupil classroom, shown in Fig. 39. The walls are brick, 12 in. thick, ceiling 12 ft. high and windows 10 ft. x 5 ft.

Solution

Exposed wall and glass $=(32+24)\times 12=672$. Exposed glass $=5\times 10=50\times 3=150$ sq. ft. Net wall $=672-150=522$ sq. ft. Wall factor $=0.30$. $(522\times 0.30+150)\times 70=21.462$ B.t.u. through wall and glass. Assume temperature of 30 deg. in attic. $24\times 32=768$ sq. ft. ceiling. Ceiling factor $=0.30$. $(768\times 0.30)=(70-30)=9216$ B.t.u. through ceiling. $21{,}462+9216=30{,}678$ total B.t.u. For low pressure, wall radiation can be taken as 300 B.t.u. $30{,}678\div 300=102$ sq. ft. of radiation. Add 10% for breakage making a total of 112 sq. ft.

Use two coils, one composed of 9 sections of 7 sq. ft. sections and one of 7 sections of 7 sq. ft., making a total of 112 sq. ft.

For 46 pupils $46\times 30=1380$ cu. ft. of air per minute for the classroom.

Assume a velocity of 400 ft. per minute in the duct. Applying formula $A=\frac{Q}{V}$. $A=\frac{1380}{400}=3.45$ sq. ft., $3.45\times 144=497$ sq. in. area of duct. Let the duct be 20 in. deep. $497\div 20$—25 in. The supply duct should be 20×25. For the resgister face $A=\frac{1380}{250}\times 144=795$ sq. in. $795\div 30=28$ in. Let the register face be $28.\times 30$ in.

CHAPTER XX

THEATRE VENTILATION

IN theatre ventilation the air for ventilating purposes is usually admitted at 70 deg., at least the system is usually designed with that in view. When the theatre is occupied, however, the air is seldom admitted at over 60 deg. to 65 deg. The heat given off by the occupants is sufficient to supply any additional heat that may be necessary. The quantity of air admitted is seldom as great as required in school work, namely, 30 cu. ft. per minute per occupant For the very best results, however, the system should be designed for that quantity. The quantity is nearer 15 to 20 cu. ft. per minute per person in the average theatre. It is sometimes determined on the basis of air change without regard to seating capacity and should always be checked up both ways. From 6 to 10 changes per hour should be provided.

Standard Method of Heating

A standard method of admitting air to the auditorium is through openings under the seats. These openings are provided with cast-iron caps called mushrooms. These mushrooms are made in three sizes, 4⅝, 5 and 6 in. in diameter and are provided with controlling dampers or adjustable tops for controlling the quantity of air from each. The velocity of air from these should be very low, not more than 200 ft. per minute as the air blows directly on the feet of the occupants and any appreciable movement of air is objectionable.

The space directly below the auditorium should be used as a plenum chamber. The fan discharges directly into this chamber and the air passes up from it through the mushrooms into the auditorium. This eliminates the necessity of any extensive system of duct work. Some systems are designed with ducts and flues leading from this plenum chamber to various parts of the building in addition to the mushrooms. This arrangement is not satisfactory, particularly if the ducts are of considerable length as only a very slight pressure can be produced in the plenum chamber owing to the large number of openings for the mushrooms, and very little air will be forced through the ducts. If this arrangement is used, the velocity in all the ducts and flues should be kept very low, not more than 300 to 400 ft. per minute.

Mushrooms are sometimes installed in the balcony floor in addition to those in the auditorium or orchestra floor. This necessitates a plenum space under the balcony which can be formed by a false ceiling. There must be large communicating ducts between this space and the plenum chamber under the orchestra floor if the same fan is used for both.

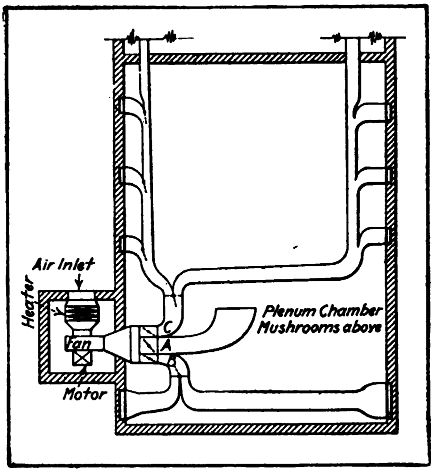


Fig. 40—Typical Arrangement of Ducts in Theatre Ventilation

If the type of fan used is what is known as the centrifugal fan, the best method to produce proper distribution of air is by means of duct work from the outlet of the fan as shown in Fig. 40. Duct A discharges into the plenum chamber under the orchestra floor. Duct B passes up to the balcony floor and discharges into the plenum space under the balcony floor. Duct C connects with the various ducts and flues leading to the register faces. Each of these three ducts is provided with a butterfly or volume damper so that the quantity of air to each may be carefully regulated. In addition to these dampers, there are also the deflecting dampers at the branch ducts.

Heating by Registers in Seats

Another method of introducing the air to the auditorium when the plenum chamber cannot be provided is by regular faces in the side of the end seats, the air blowing directly into the aisle as shown in Fig. 41. This method is particularly adaptable in churches, where the seats are continuous and of the bench type.

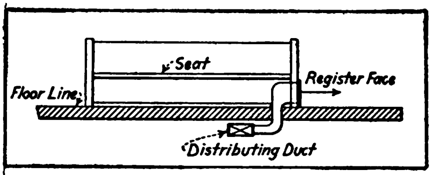


FIG. 41—REGISTERS IN END OF SEATS DELIVERING AIR TO AISLES

Exhaust fans are not always installed in theatre work, but if good results are desired they should be used. If they are used it is customary to install one large disc fan in the space which is usually provided above the false ceiling over the auditorium. Openings can be provided into this space through the ornamental ceiling. The fan then draws the air from this space which is continually replaced by the air in the auditorium and discharges it through the roof. Some form of ventilator or vent cap should be provided for the opening in the roof.

If there are toilet rooms in the basement or in the interior of the building without outside exposure, these should have a separate exhaust system. An air change of from 6 to 10 times per hour should be provided for those rooms to ensure thorough ventilation.

CHAPTER XXI

HOTEL VENTILATION

THE problems encountered in the ventilation of large hotels are probably the most difficult of any class of buildings and the portion of the hotel which must be handled with particular care is the kitchen. The kitchen is usually located in that portion of the building which is entirely below the ground level, with no outside exposure and because of the large amount of cooking done and the intense heat generated from the ranges, broilers, etc., it is very difficult to prevent odors from reaching other portions of the building.

The dining rooms and restaurants are usually directly over the kitchen, and the natural tendency of the odors is to rise. In order to eliminate this, a plenum condition must be produced in the dining rooms, restaurants, lobbies, and all rooms from which odors must be kept and which are in any way connected with the kitchen or other rooms where objectionable odors may arise. This is accomplished by exhausting with the exhaust fans only about 60 to 70 per cent as much air as is blown in by the supply fans. In the kitchen and similar rooms a vacuum condition must be produced by exhausting at least 50 per cent more air by means of exhaust fans than is blown in by the supply fans. These unbalanced conditions in the two portions of the building produce a steady current of air flowing from the upper rooms down through doors and connecting passageways to the kitchen and out through exhaust ducts. With the various systems properly balanced, this arrangement will prevent any odors reaching any other rooms than the ones in which they are generated.

Kitchen Ventilation

The ventilation of the ranges and all cooking apparatus must be handled by an entirely separate system. Each range, broiler, vegetable steamer, etc., must be provided with a separate hood over the top which entirely covers the space above the apparatus. An exhaust duct should be taken from the top of each hood, and these ducts combine into one main flue leading to the roof. A centrifugal exhaust fan for this flue should be installed on the roof and should discharge through a copper vent cap. A bypass should be provided around the fan or a duplicate exhausting apparatus be furnished to make the system absolutely reliable.

The duct work for this system should be constructed of heavy black iron not less than No. 12 gauge in thickness. These exhaust ducts and flues should also be covered with 85 per cent magnesia blocks not less than 1 in. thick and finished with ½-in. hard cement. The vaporized oils and fat generated in cooking are drawn up into the exhaust ducts and condense, forming a very inflammable coating on the inside. This is liable to become ignited and burn with disastrous results if every precaution is not taken to guard against it. An automatic fire damper with a fusible link so arranged as to close in case of fire on the inside, should be provided in the main exhaust duct near the point where the last branch duct joins the main duct. Clean-out doors should also be provided at accessible points so that the interior can be cleaned at frequent intervals.

Range Hoods

The hoods which are placed over the ranges, broilers, etc., are termed "range hoods." These are constructed of sheet metal in the form of a cap, and should be of sufficient size to cover the entire range. They should be located as near to the range as possible to give ample working space. The outside lower edge should be about 6 ft. 6 in. from the floor. The usual method of constructing these hoods is shown in Fig. 42. There may be one or more openings through the top into the exhaust duct depending on the size of the hood. It is sometimes found

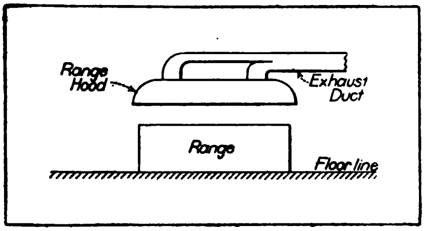


Fig. 42.—Ordinary Single Range Hood

advantageous to place several ranges together in one group. When this is done, one large hood will serve for the entire group, with a corresponding number of openings in the top to the exhaust duct. With this type of hood, it will be found that at certain times even with a good flow of air out through the exhaust duct that some of the fumes and smoke arising from the cooking will escape from under the hood. One means of eliminating this is to construct a double hood as shown in section in Fig. 43.

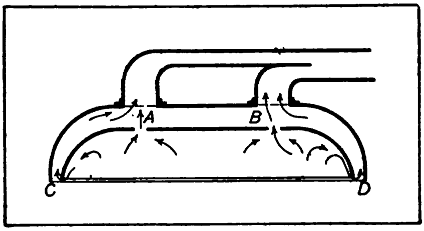


FIG. 43.—DOUBLE RANGE HOOD

Within the hood is an inner casing spaced a short distance from the outer hood. This forms a chamber from which the exhaust duct draws the air. There are openings through the top of the inner casing directly below where the exhaust duct enters the outer hood shown at A and B. The two hoods are not joined at the lower outside edge which leaves another opening into the chamber around the entire outside edge of the hood shown at C and D in the illustration. About one-half of the air carried away by the exhaust duct should pass through the openings in the top and the remainder through the opening around the edge. By proportioning these areas properly, perfect results can be obtained. Any gases and smoke not taken out through the openings in the top which tend to escape from under the hood spread out in a thin film and, passing out close to the edge, are caught by the inrush of air at this point and effectively removed.

Quantity of Air for Range Hoods

To estimate the exact quantity of air that should be removed through the range hoods is a difficult matter, as there are no formulas which apply to this and no very reliable mathematical method of solution.

One empirical method given is to allow 1 sq. ft. of exhaust duct for each range. This is very indefinite, however, as nothing is stated as to the velocity of air in the exhaust duct.

Allowing a velocity of 1200 ft. per minute in the main exhaust duct gives 1200 cu. ft. of air per minute for each range. This rule must be modified, however, to suit conditions. A better method perhaps is first to locate the hoods on the plans to conform with the kitchen layout. Locate the additional exhaust registers in the various portions of the kitchen that may be necessary. Then estimate the size of the range hood, exhaust ducts and flues on an air change basis, with reference to the volume of the kitchen. The largest proportion of the air should be removed through the range hood vents. This proportion can be determined from the number and location of the additional exhaust registers. The supply registers should be carefully located with reference to exhaust registers and the range hoods so as to produce a uniform distribution of air.

The quantity of air exhausted shou'd be sufficient to produce an air change about every 3 minutes, or twenty air changes per hour. The amount of fresh air supplied should be about 60 per cent to 70 per cent of this.

The velocities assumed for proportioning both the supply and exhaust ducts should be about the same as given for school ventilation. These velocities might be exceeded slightly where large quantities of air are handled. The maximum velocity in the main supply and exhaust ducts should not exceed 1200 ft. per minute and in the vertical flues to registers, 600 ft. per minute. The main exhaust flue to the roof should be proportioned for a velocity of about 1200 ft. per minute.

In the boiler and engine rooms which are as a rule entirely below the ground level with no outside exposure from 20 to 30 air changes per hour should be provided. Both the supply and exhaust system should be separate from the other units.

The main dining rooms are usually very large in volume with high ceilings, and as a rule can be handled most satisfactorily with the indirect or hot blast system. Because of the comparatively small amount of exposed wall and glass surface the temperature of the entering air can be kept low and practically eliminate the objectionable features of this system. The system when properly designed is very flexible and the room temperature can be very easily controlled under varying conditions

A complete system of automatic temperature control should be provided. The arrangements of ducts, heaters and special devices for automatic temperature control will be fully explained in the succeeding chapters. The general scheme of introducing the air at or near the ceiling and exhausting near the floor line as already outlined can be adopted with satisfactory results. Often the ornamental design of the walls of the room make it objectionable to put registers in them. If this is the case, it is possible to use concealed openings behind a cove at the ceiling line. This makes a very good arrangement, particularly if the ceiling is curved or dome shaped. No register faces are necessary, and it is advisable to introduce the air at a comparatively high velocity, say 500 to 600 ft. per minute. The velocity carries the air well into the center of the room and produces good distribution. The exhaust registers can be long and narrow in design and located conveniently in the base board and their appearance is not objectionable.

Register Sizes

When either the supply or exhaust registers are located within 8 ft. of the floor line, the velocity of the air leaving or entering should not exceed 300 ft. per minute, and in determining the sizes of the registers the net free area should be used. The free area, through register openings, will vary from 50 per cent. to 80 per cent. of the total area, depending upon the design. The percentage of free area for each particular design is usually given in the manufacturers' catalogs. When the size of the register is given it refers to the open portion of the register or the opening required in the wall and not to the outside dimensions. The outside dimensions are from ¾ in. to 1½ in. more on each side. Assume that the amount of air to be discharged from a particular register is 800 cu. ft. per minute. The velocity of discharge to be 250 ft. per minute. Then $\frac{800}{250}=3.2$ sq. ft. net area required, or $3.2 \times 144 = 460$ sq. in. Assume the particular design of register face selected has a net free area of 60 per cent:

Let A equal the total area of the register. Then 60 per cent. of A must equal 460 sq. in., or

$$A \times 0.6 = 460 \qquad A = \frac{460}{0.6} = 767 \text{ sq. in.}$$

Let the register be 20 in. deep.

Then $\frac{767}{20} = 38.3$, say 38 in.

The register should be 20 x 38 in.

If there is a 1-in. flange or border the outside dimension of the face would be 22 x 40 in.

Reversing Dampers

In all such rooms as banquet halls, ball rooms, etc., which are liable to become over heated when there is a large assembly or where there may be an excessive amount of smoking, a reversing damper connected with the main supply and exhaust duct should be provided. These should also be provided where an air cooling system is installed in connection with the ventilating system for summer use.

The idea of the reversing damper is to reverse the direction of flow at the outlets to the room. With the reversing damper set in one position the fresh air enters at the ceiling and the foul air is exhausted at the floor line. With the damper reversed, the fresh air is admitted at the floor line and the foul air exhausted at the ceiling. One form of reversing damper which is very satisfactory is shown in Fig. 44.

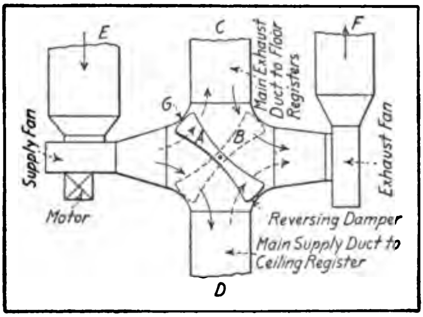

E
C
F
Main Exhaust Duct to Floor Registers
G
A
B
Supply Fan
Exhaust Fan
Motor
Reversing Damper
Main Supply Duct to Ceiling Register
D

FIG. 44.—SATISFACTORY FORM OF REVERSING DAMPER

For normal operation the reversing damper is set as shown by the full lines, position A. The supply fan then draws the air from duct E and forces it into duct D, which leads to the register faces at the ceiling. Duct C is connected with the register faces located at the floor line. The foul air passing through the exhaust registers flows to the main exhaust duct C and to the exhaust fan.

If it is desired to reverse the direction of flow or to blow fresh air in at the floor line and exhaust from the ceiling, the damper is set in position B as shown by the dotted lines. The air then comes in from the fresh air inlet through duct E to the supply fan the same as before. Leaving the supply fan, however, it now flows in the opposite direction as indicated by the dotted arrows to duct C and to the registers at the floor line. The foul air, drawn by the exhaust fan flows out at the ceiling registers back through duct D in the direction indicated by the dotted arrows to the exhaust fan and out through duct F. At the point G on the reversing damper should be provided a heavy rubber and canvas to form an air tight joint between the damper and the casing and still be flexible enough to permit the damper to be easily moved. This projection should also extend across the top and bottom of the damper. The damper should be pivoted at the center and balanced very carefully. On small sizes the damper can be operated by means of a lever from the outside. On larger sizes, however, it is better to provide entrance doors at convenient places in the ducts and operate the damper from the inside. Some means of locking the damper in position should be provided.

If this arrangement is applied to the system with a complete equipment of thermostatically operated mixing dampers and individual ducts to each room, then the reheating stack with its by-pass and mixing dampers must be placed beyond the reversing damper in duct D. Otherwise the reheating stack may be placed back of the supply fan in duct E.

As to the principle of introducing the air at the floor line or at the ceiling, a general statement might be made to the effect that whenever the air entering the room is below the room temperature the air should enter at the floor line and be exhausted at the ceiling line. Whenever the air entering the room is above the room temperature the air should enter at the ceiling and be exhausted at the floor line. The flow of air is then at all times in the direction of natural flow or with the force of gravity. Assume that the air is entering at a temperature below that of the room, as would be the case with an air cooling system in operation in the summer time, or at such times when a room had become overheated; as the air enters the room its temperature is raised and it becomes lighter. Its natural tendency, therefore, is to rise and the logical way to handle the air under these conditions is to introduce it at the floor line and exhaust it at the ceiling, thus allowing the force of gravity to assist in the movement of the air rather than to oppose it.

When the air enters at a temperature higher than that of the room the movement is in the opposite direction. The air after entering is lowered in temperature and becomes heavier; therefore, its tendency is to fall. The air under these conditions should be blown in at the ceiling and exhausted at the floor. Therefore, in order to have a system applicable to all conditions it should be equipped with a reversing damper. For winter service the damper would normally be set in position A and for summer serivce in position B.

Fresh Air Intake

The fresh air intake should be located preferably on some exterior part of the building and never on an interior court unless the court has a very large area. If there is any appreciable downward velocity in the court caused by the suction of the fan, part of the air will be drawn from any open windows facing on the court. In mild weather there will be a large number of open windows facing on the court and the greater portion of the air will be drawn from within the building. This air will be at a somewhat higher temperature than the outside air and may result in overheating of the rooms supplied. The roof of the building is a good location for the fresh air intake, but of course this requires additional power to draw the air from such a distance. It should always be located at a considerable distance from the stack and from any exhaust outlets.

Toilet Room Ventilation

The toilet rooms on all typical floors are usually located in the same relative position on each floor so as to bring all the piping together, a vent shaft being provided in which to run all this piping. This shaft forms a very good means of providing ventilation for the toilet rooms. Each room should be provided with a register face of proper size for the amount of air to be removed. It is not necessary to construct a galvanized iron flue within the shaft as the shaft itself may be used as the flue.

A short galvanized iron connection should be provided, however, for each register face as shown in Fig. 45.

The duct should extend up about 3 ft. and a volume damper be furnished with set screw to lock it in any desired position. This arrangement prevents any tendency of the air from one room passing out into another room. It also provides a means of regulating the flow from each room. The top of the shaft should be connected with an exhaust fan and the discharge carried through a vent cap on the roof.

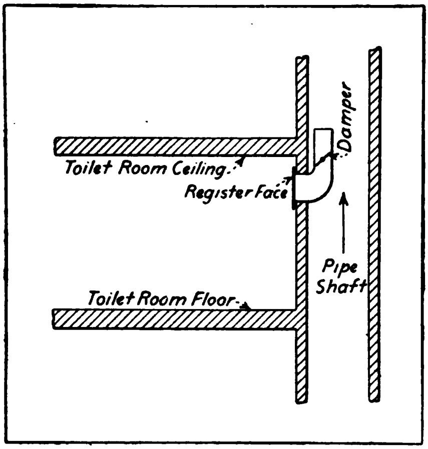


FIG. 45.—METHOD OF CONNECTING AIR FACES TO VENT SHAFTS

CHAPTER XXII

HEATING stacks or heaters, as they are more properly called when used in connection with ventilating systems are constructed in two different methods, by pipe coils and by cast-iron surfaces. Pipe coil heaters are usually constructed of 1-in. pipe assembled as shown in Fig. 46. The heater illustrated is formed of four different groups or sections, each section being made of four rows of pipe screwed into a cast-iron base or header. The pipes should be spaced on 2¾-in. centers and the successive rows staggered, so as to expose a maximum amount of heating surface to the air passing through. The temperature to which the air passing through will be heated is dependent upon four things, namely, the temperature at

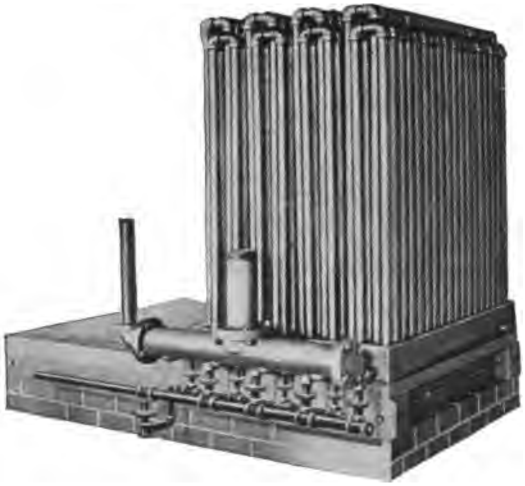


FIG. 46—FOUR-SECTION PIPE COIL HEATER, CONSTRUCTED OF ONE-INCH PIPE

which the air enters the heaters, the velocity with which it passes through, the temperature of the steam or water within the coils, and the number of sections through which the air passes. The more rapid the velocity of air through the heaters the lower will be the final temperature, but the rate of condensation of steam or the rate of transmission of heat through the coil surfaces increases with the velocity.

To determine the rise in temperature through the heater under various conditions, the following formula may be used:

$$R = \frac{T - t_o}{K} \qquad (1)$$

T = Steam temperature.

t_o = Temperature of incoming air.

R = Rise in temperature of air.

K = Constant which is determined from table XXV.

If the velocity through the heater is known and also the number of sections, K can be determined from the table, and the rise in temperature of air determined by substituting these values in the formula.

To illustrate, assume four sections deep with a velocity of 1200 ft. per minute through the heater, steam pressure 5 lb., $T = 227$ deg. and air entering at zero, $t = 0$.

TABLE XXV

K is as Follows for any Pressure and Initial Temperature

No. of Sections Deep	300 ft. Vel.	450ft. Vel.	600 ft. Vel.	900ft. Vel.	1200 ft. Vel.	1500 ft. Vel.	1800 ft. Vel.	2100 ft. Vel.	2400 ft. Vel.	3000 ft. Vel.
1	3.90	4.46	4.91	5.57	6.20	6.66	7.09	7.45	7.80	8.40
2	2.19	2.50	2.76	3.13	3.48	3.75	3.97	4.19	4.38	4.71
3	1.62	1.85	2.04	2.30	2.56	2.75	2.92	3.08	3.22	3.48
4	1.33	1.52	1.68	1.91	2.12	2.28	2.42	2.55	2.67	2.87
5	1.21	1.35	1.46	1.66	1.85	1.99	2.11	2.22	2.32	2.50
6	1.14	1.23	1.32	1.49	1.66	1.78	1.90	2.00	2.08	2.26
7	1.11	1.06	1.24	1.38	1.54	1.66	1.76	1.85	1.94	2.08
8	1.09	1.13	1.19	1.31	1.44	1.55	1.65	1.73	1.81	1.95
9	1.07	1.11	1.15	1.26	1.36	1.46	1.55	1.64	1.71	1.85
10	1.06	1.10	1.13	1.22	1.30	1.40	1.49	1.57	1.64	1.77

Referring to the table, for four sections deep and a velocity of 1200 ft. per minute, $K = 2.12$.

$$R = \frac{227 - 0}{2.12} = \frac{227}{2.12} = 107 \text{ deg.}$$

Final temperature of air will be 107 deg.

Assume that the air under the above conditions enters at 10 degrees below zero, instead of zero t then equals (—10).

$$R=\frac{227-(-10)}{2.12}=\frac{227+10}{2.12}=\frac{237}{2.12}=111.8 \text{ deg.}$$

The total rise in temperature is 111.8 deg. The air enters at —10 deg. Therefore the final temperature is 101.8 deg.

The same formula may be used to determine the number of sections to use if a certain rise in temperature of the air is required. The formula should then be solved for K and expressed as follows:

$$K=\frac{T-t_o}{R} \qquad (2)$$

Let it be required to obtain a temperature rise of 85 deg. through the heater with air entering at zero degrees. Determine the number of sections and the velocity of air to use, temperature of the entering air zero degrees, and steam temperature 227 deg.

$$K=\frac{227-0}{85}=\frac{227}{85}=2.67$$

Referring to the table nearest constant to this is 2.56, with 3 sections, and a velocity of 1200 ft. per minute. Taking this constant 2.56 and solving for R, gives a rise of 88.8 deg., which would be near enough for practical purposes and also give a slight margin of safety.

After the number of sections has been determined, the next step is to determine the size of the sections and number of square feet of surface. Divide the total quantity of air in cu. ft. per min. by the velocity in ft. per min. and the result is the square feet of free area through each section. With the square feet of free area, the size of the section can be selected from the manufacturers' catalogues. With pipes on 2¾ in. centers the free area in square feet can be obtained by dividing the lineal feet of pipe by the number of pipes deep and this by 8.4. It is advisable, however, to also give the square feet of heating surface which each section contains. In order to estimate this, the average rate of transmission in B. t. u. per square feet of heating surface per hour must first be obtained. For this purpose the following formula taken from the American Blower Co., catalogue may be used:

$$H = C\sqrt{V}(t_s - t_o) \qquad (3)$$

V = Velocity in feet per second.

t_s = Temperature of steam.

t_o = Temperature of entering air.

H = B. t. u. per square foot of surface per hour.

C is a constant for different numbers of sections is given in table XXVI.

TABLE XXVI

No. of Sections	1	2	3	4	5	6	7	8
C =	3.45	3.00	2.65	2.33	2.12	1.95	1.80	1.65

Assuming a steam temperature of 227 deg. and an entering air temperature of 0 deg., table XXVII gives the values of H from formula (3) for four different velocities:

TABLE XXVII

Velocity of Air in feet per Minute	No. of Sections Deep 1	2	3	4	5	6	7	8
800	2840	2470	2164	1920	1750	1606	1450	1360
1000	3200	2790	2440	2170	1900	1810	1670	1535
1200	3500	3040	2670	2360	2150	1980	1825	1678
1500	3950	3400	2981	2645	2400	2220	2020	1870

With these tables it is quite possible to determine the B. t. u. transmission for each square foot of surface under all conditions. If the total B. t. u. required to heat the air from the entering temperature to the final temperature is determined and this quantity divided by the B. t. u. per square foot, the result will be the square feet of heating surface necessary. The formula for this has already been given in preceding chapters as follows:

$$\text{B. t. u.} = \frac{Q \times T}{55}$$

B. t. u. is the total B. t. u. required and must be per hour. Therefore, Q, the cubic feet of air, must be expressed in cubic feet per hour. T is rise in temperature. This gives the total

B. t. u. necessary to heat the air. This quantity should also be used in determining the size of supply and return mains to the heater, the boiler capacity, etc., as will be explained more fully in succeeding chapters.

If there are three or four sections in the heater, as is usually the case, each section does not condense the same amount of steam. The first section, or the coldest section, naturally condenses the most, being in contact with the coldest air, and as the air is heated nearer to the temperature of the steam, the rate of condensation naturally decreases. To find the condensation in each section, or the B. t. u. given up by each section, the rise in temperature for each section must be determined by the use of Formula 1 and Table XVII. The rise in temperature for the first section is determined by taking the constant from the table for one section deep and for the velocity selected. The rise in temperature for the second section is then obtained by using the entering temperature (t_o) of the second section as the final temperature of the first section and so on until each section is obtained.

Problem

Find the number of sections, total square feet of surface, free area through sections and size of sections necessary for the problem in Chapter XVII.

The quantity of air is 840,000 cu. ft. per hour or 14,000 cu. ft. per minute. The necessary temperature of air entering the room is 131 deg. Assume the air to be heated to 135 deg. in the heater.

Steam temperature, 227 deg.

Entering air temperature, 0 deg.

Using formula (2).

$$K=\frac{T-t_o}{R}$$

$$=\frac{227-0}{135}=1.68$$

This rise can be obtained by using six sections with a velocity of 1200 ft. per minute through the heater, or seven sections with a velocity of 1500 ft. per minute. As the resistance would be less with the lower velocity, use six sections with 1200 ft.

velocity. Free area through each section $=14{,}000 \div 1200 = 11.66$ sq. ft. Total B. t. u. per hour required is

$$\frac{840{,}000 \times 135}{55} = 2{,}062{,}000.$$

Referring to Table XX, with six sections and a velocity of 1200 ft. per minute, the B. t. u. per square foot is 1980.

$$\frac{2{,}062{,}000}{1980} = 1040 \text{ sq. ft. total heating surface required.}$$

$$1040 \div 6 = 173.3 \text{ sq. ft. per section.}$$

To determine the temperature rise through each section, referring to Table XXV, with one section and a velocity of 1200 ft. per minute:

$$K = 6.2.$$

$$R = \frac{227-0}{6.2} = 36.6 \text{ deg.}$$

For the temperature rise in the second section:

$$t_o = 36.6.$$

$$R = \frac{227-36.6 \text{ deg.}}{6.2} = 30.7 \text{ deg.}$$

For the third section:

$$t_o = 36.6 + 30.7 = 67.3 \text{ deg.}$$

$$R = \frac{227-67.3}{6.2} = 25.7 \text{ deg.}$$

The succeeding sections are figured in the same way. The B. t. u. supplied and consequently the steam condensed by each section, is figured as before explained from the total quantity of air and the temperature rise for each.

CHAPTER XXIII

VENTO HEATERS

THE recent development of cast-iron heaters has gradually replaced the pipe coil heaters, and, at the present time, cast-iron heaters of the Vento type are largely used for indirect heating in all classes of buildings. On the outer surface of each unit are cast projections. These projections are hollow, extend at right angles to the direction of flow of the air, and thus present a very thin surface for the effective transmission of heat from the steam to the air. Fig. 47 shows a cross-section of one

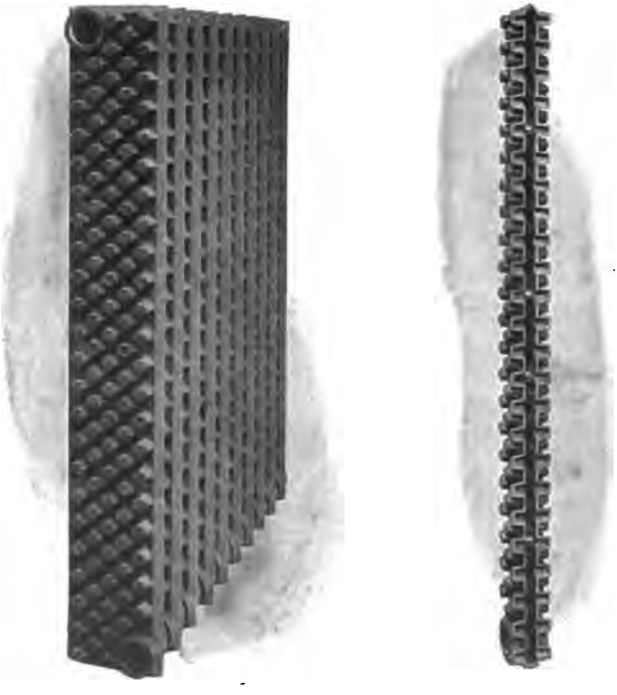


FIG. 47—SIDE VIEW AND CROSS-SECTION

loop and a side view of a section composed of ten loops. The loops are made in standard widths of 9⅛ in. and narrow widths of 6¾ in. Each of these is made in heights of 40 in., 50 in. and 60 in., the standard widths having an exposed surface of 10.75 sq. ft., 13.5 sq. ft. and 16 sq. ft. per loop and the narrow width 7.5 sq. ft., 9.5 sq. ft. and 11 sq. ft. respectively. There is also a 30-in. regular section containing 8 sq. ft. per section. These sections are assembled with the loops on 4 in., 4⅝ in., 5 in. and 5⅜ in. centers. Table No. XXVIII gives the net square feet of free area per loop for these spacings and for the different heights.

TABLE XXVIII

Height of Loops	Spacing			
	4 in.	4⅝ in.	5 in.	5⅜ in.
30 in..........	.255	.390	.460	.542
40 in..........	.350	.525	.620	.729
50 in..........	.428	.650	.768	.905
60 in..........	.511	.781	.921	1.08

The net air space in square feet through the heater is determined in the same manner as has been explained for pipe coil heaters. The velocity of air passing through the heater is first determined. The total quantity of air in cubic feet per minute divided by the velocity in feet per minute gives the net cross-section area for each section. From this net area the number of loops required per section can be determined from the above table. Which of the four heights to use is determined from the local conditions. The manufacturers' catalogues give tables for determining the temperature rise through the stacks. These tables are arranged for velocities from 200 ft. per minute up to 1800 ft. per minute, for the four different spacings given above for groups of from 1 to 8 stacks deep and also for various temperatures of steam or water in the stacks. With these various tables any desirable combinations can be made and any condition met.

To illustrate the application of these tables, a portion is given in table No. XXIX for temperature rise with loops spaced on 5-in. centers, with air entering at 0 deg., 20 deg. and 30 deg., with velocities through the coils from 800 to 1400 and one to five sections deep. The steam pressure is 5 lb., 227 deg. Fahr. temperature. The letters F. T. in the table refer to final temperatures. The column C gives the pounds of steam condensed per square foot of surface per hour.

The first thing that must be known in proportioning an indirect stack is the necessary temperature rise of the air. This is determined in the manner already described, from the heat losses in the rooms, or from zero to 70 deg. if ventilation only is required. Knowing the required temperature rise the approximate velocity through the heater must be selected and this is dependent upon the type of fan which is to be used. With the centrifugal type, which is most generally used, except when the duct work is very simple with short straight runs, a velocity of about 1200 ft. per minute can be used. If a disk type of fan is used, the velocity through the heater should not exceed 600 ft. per minute because of the increased resistance at the higher velocities. When the approximate velocity has been assumed from the table of final temperatures under this velocity, the

TABLE XXIX

No of Stacks Deep	Temperature of Entering Air	Velocity Through Heater in Feet Per Minute							
		800		1000		1200		1400	
		F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.
1	0	38	1.95	35	2.24	32	2.46	30	2.65
	20	54	1.75	51	1.99	49	2.23	47	2.42
	30	62	1.64	60	1.92	58	2.17	56	2.33
2	0	68	1.74	62	1.99	58	2.23	54	2.42
	20	81	1.57	76	1.80	72	2.00	69	2.20
	30	87	1.46	83	1.70	79	1.89	76	2.06
3	0	93	1.59	86	1.84	81	2.08	76	2.27
	20	103	1.42	97	1.65	92	1.85	88	2.06
	30	108	1.33	103	1.56	98	1.75	94	1.91
4	0	113	1.45	106	1.70	100	1.92	95	2.13
	20	122	1.31	115	1.52	110	1.73	105	1.91
	30	126	1.23	120	1.44	115	1.63	110	1.80
5	0	129	1.32	122	1.56	115	1.77	109	1.96
	20	136	1.19	130	1.41	124	1.60	119	1.78
	30	140	1.13	134	1.33	128	1.51	123	1.67
6	0	143	1.22	135	1.44	129	1.65	123	1.84
	20	148	1.10	142	1.30	136	1.49	130	1.65
	30	151	1.04	145	1.23	139	1.40	134	1.56
7	0	154	1.13	147	1.35	140	1.54	135	1.73
	20	159	1.02	152	1.21	146	1.39	141	1.55
	30	161	.96	155	1.15	149	1.31	144	1.46

nearest temperature to the one desired is found, and opposite this is found the number of stacks deep the heater should be.

Applying this to the problem given in Chapter XVII, the quantity of air per hour was 840,000 cu. ft. which is equal to 14,000 cu. ft. per minute. The desired temperature rise was 131 deg. Let the required temperature of the air leaving the stacks be 135 deg. the same as was assumed with pipe coils. Referring to the temperature chart, under the 1400-ft. velocity column is found, with zero entering temperature, a final temperature of 135 deg. with a heater seven sections deep. This system being designed for factory work, a velocity of 1400 ft. per minute through the heater could be safely used. The heater therefore should be seven stacks deep.

The quantity of air (Q) is 14,000 cu. ft. per minute, the velocity (V) is 1400 ft. per minute, therefore, the square feet of free area through the stack should be $\frac{14{,}000}{1400} = 10$ sq. ft. for each section.

Let it be assumed that the 60-in. high sections will be best suited for the space conditions given. Referring to the table of free areas for the 60-in. sections on 5-in. centers, the free area per section is found to be 0.921 sq. ft. per loop. Therefore, $\frac{10}{0.921} = 10.8$ loops. Using the next whole number each section should contain 11 loops. The distance between the center of the loops being 5 in., the total width of each section will be 55 in. In setting up the heaters, the sections should be staggered, that is, the centers of the loops in one section should be placed opposite the openings between the loops in the next section. For the total width of the casing, 2½ in. should be added to the width of the section for staggering, making the casing 57½ in. wide.

The 60-in. regular section contains 16 sq. ft. of surface per loop. Total surface per section is 16 x 11 = 176 sq. ft. In the entire heater there should be 176 x 7 = 1232 sq. ft.

Referring again to the final temperature table, under the condensation column corresponding to the temperature rise of 135 deg. with a velocity of 1400 ft. per minute and seven stacks deep, the average condensation per square foot of surface per hour is found to be 1.73. As there is a total of 1232 sq. ft. of surface, the total condensation in the heater will be 1232 x 1.73 = 2132 lb. of steam per hour. From the steam tables it is found that the latent heat of steam at 5-lb. pressure is 955 B.t.u. Assuming that the water of condensation leaves the stacks 10 deg. below the temperature of steam, the total B.t.u. per pound of steam condensed would be 965. The total heat given up by the heater would be 2132 x 965 = 2,057,400 B.t.u. per hour.

This could have been determined directly from the quantity of air, as already explained. To check the above result the quantity of air per hour is 840,000. The temperature rise is 135 deg. Therefore, $\frac{840{,}000 \times 135}{55} = 2{,}062{,}000$ B.t.u. per hour.

To estimate the necessary boiler capacity for supplying heater stacks, the total amount of steam condensed must be determined as above described and the size of the boiler, grate surface, etc., is estimated in the same manner as for direct radiation described in Chapter XIV. If the system is a combination of direct radiation and indirect, the steam required by the two parts of the system is determined, and the sum of the two is the total amount of steam that must be supplied. The necessary boiler capacity for the indirect can also be determined directly from the quantity of air and the temperature rise by determining the B.t.u. that must be supplied.

The size of steam and return mains can be obtained in the same manner from the pounds of steam supplied. A better method, however, is to express the amount of indirect surface in equivalent amount of direct surface, as the tables given for sizes of steam mains are given in terms of the amount of direct surface which they will supply. The ratio between the indirect and equivalent amount of direct may be obtained from the B.t.u. basis or from the comparative rates of condensation. The tables for the capacities of steam mains already given in Chapter IX and XI are based on the assumption that each square foot of direct radiation gives off 250 B.t.u. per hour and condenses 0.25 lb. of steam per hour. If (R) represents the rate of condensation, then $\frac{R}{0.25}$ is the number by which to multiply the square feet of indirect surface to obtain its equivalent amount of direct surface. If the square feet of indirect surface in the heater is not known, the equivalent amount of direct surface may be determined from the amount of air and the temperature rise by first obtaining the total number of B. t. u. that must be supplied per hour. As each square foot of direct surface supplies 250 B.t.u. per hour, this quantity divided by 250 gives the equivalent direct radiation. Expressing this in the form of an equation, let Q = number of cubic feet of air per minute, then $Q \times 60$ = number of cubic feet per hour. Let T = temperature rise, in degrees Fahr., of the air passing through the heater, then $\frac{Q \times 60 \times T}{55 \times 250}$ = equivalent direct surface. When the equivalent surface has been determined, the size of steam and return mains may be obtained in the same manner as for direct radiation. The rate of condensation should be estimated for each section of the heater separately in order to get the proper size of each connection.

Problem

Determine the size boiler, the size of steam and return main and size of individual connections to heater referred to above.

The heater consists of seven sections, each section containing 176 sq. ft. of surface. The average rate of condensation was 1.73 lb. per square foot per hour. Therefore the steam condensed per section is $176 \times 1.73 = 305$ lb. per hour. Total steam is $305 \times 7 = 2135$ lb. per hour. $\frac{2135}{34.5} = 62$ boiler horse-power necessary. To find the heating surface, refer to Chapter XIV $\frac{2135}{3} = 712$ sq. ft. heating surface. To determine the size of grate the height of the stack should be known. Assume a rate of 12 lb. coal per square foot of grate, then, $\frac{2135}{8 \times 12} = 22.2$ sq. ft. of grate surface.

To determine size of supply main to heater the distance from boiler to heater should be known. Assume the distance less than 100 ft. The rate of condensation in the heater is 1.73. $\frac{1.73}{0.25} = 6.92$, the ratio between the actual surface and the equivalent direct surface. Total actual surface in the heater is 1232 sq. ft. Therefore, $1232 \times 6.92 = 8526$ sq. ft. $\frac{8526}{7} = 1218$ sq. ft. in each section. Referring to table of steam mains in Chapter IX for a two-pipe gravity system, the supply main should be 7 in. in diameter and the return main 4 in. The connections to each section should be 3½ in. and 2½ in.

CHAPTER XXIV

ARRANGEMENT OF HEATERS

FOR factory work the sections are usually placed together in one group. The sections are vertical with the steam connections at the top and return connections at the bottom. Each section should be valved separately so that in mild weather one or more sections may be used to obtain the desired air temperature. The heater should be encased on all four sides with a sheet metal casing. This is usually constructed of black iron and should be at least No. 14 gauge. If an air washer is used in connection with the system, the heater should be divided into two units. The first unit, consisting of about two sections, is called the tempering coil. The second unit is made up of the remaining sections, and is called the reheater. The air washer should be placed between the two groups. The temperature rise should then be determined for each group separately. The tempering coil should be so proportioned as to give a temperature of about 50 deg. entering the air washer. This eliminates any danger of the water in the washer freezing. To determine the number of sections for the reheater a drop of about 10 deg. should be assumed for the air passing through the washer, which would make the temperature of air entering the reheater 40 deg. The number of stacks deep for the reheater would then be determined in the same manner as described, with the entering temperature 40 deg., instead of 0 deg.

A by-pass should be provided below both the tempering coil and reheater. The area through the by-pass should be such that at least 75 per cent. of the air will pass through when the damper is wide open.

For Schools and Theatres

The standard of arrangement vento heaters for schools and theaters where the air is to be introduced at 70 deg. is three stacks in one group of regular vento, the loops on 5-in. centers and the velocity through the stacks about 1200 ft. per minute. This gives a possible temperature rise of 80 deg. in

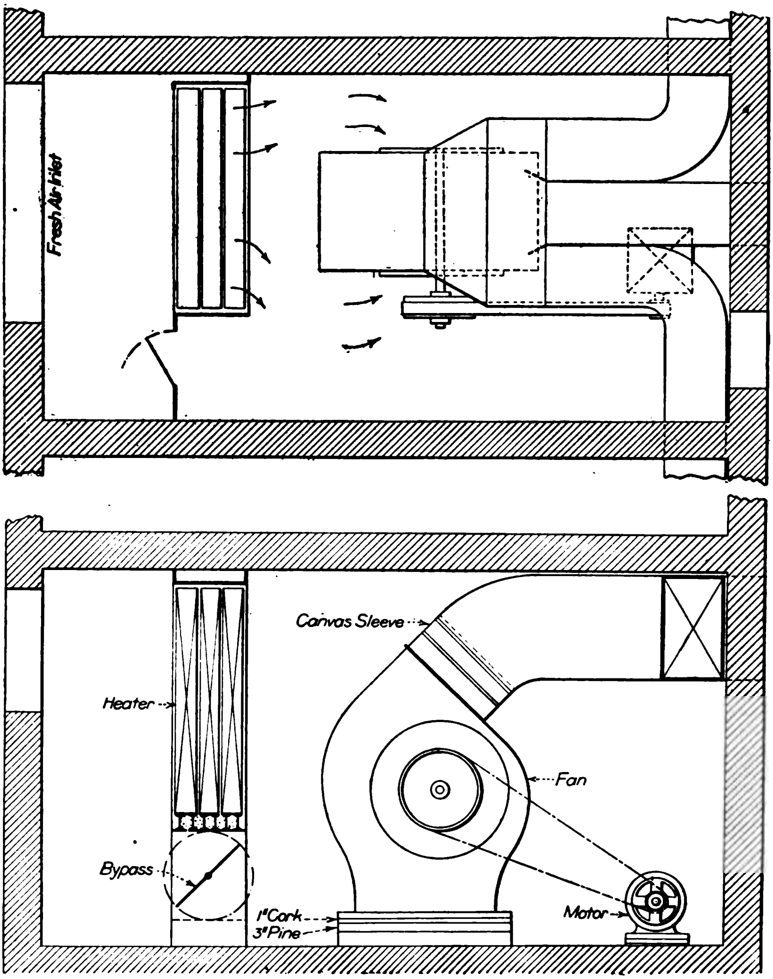


Fig. 48—Arrangement of Heaters and Fans for School Ventilation.

zero weather, which allows a little margin of safety for imperfect air removal from the stacks, etc. A by-pass of at least 75 per cent. of the net free area through the stacks should be provided either above or below the heater. The damper in this by-pass can be either controlled by hand or by a thermostat. Fig. 48 shows a plan and elevation of a typical arrangement.

In small schools or buildings where a gravity return system of steam circulation is desired, it may happen that there will not be sufficient head room to place the stack vertically in the usual manner and have the base of the stack come far enough above the water line of the boiler to provide proper return. The bottom of the stack or the return connection should be at least 24 in. above the water line of the boiler. It would be better to keep this distance 30 to 36 in. if possible. If this condition is encountered the difficulty can usually be overcome by hanging the stacks in a horizontal position with the air passing either up or down through them in a vertical direction. It is better to design the system so that the air passes up if possible and the movement is then with the force of gravity. The stacks should never be placed on the side with the air passing through horizontally, as the pockets in the projections would immediately fill with water and reduce the amount of heat transmission.

By referring to the vento catalogue it will be found that, with one section deep and the loops on 4-in. centers, the temperature rise for velocities of 400, 500 and 600 ft. per minute are respectively 80, 74 and 70. It can therefore be seen that when the head room is such as to permit the use of only one stack placed in a horizontal position, the required conditions can be obtained by having the heater made up with the loops on 4-in. centers. It would be advisable under these conditions to determine the free area and size of stack for a velocity of 400 ft. per minute.

Fig. 49 shows a typical arrangement for a system designed in this manner and applicable to small schools having not over four or six class rooms. If this scheme were to be used on larger schools, it would be advisable to divide the school into groups of about four class rooms each and install a separate unit, as shown, for each group. It is doubtful, however, if the first cost of installation would be decreased to any extent over a central distributing system, unless the layout were such as to save considerable work. The heater shown in Fig. 49 should be divided into at least two sections so as to provide means of control, and it would be advisable to provide thermostatic control of one section.

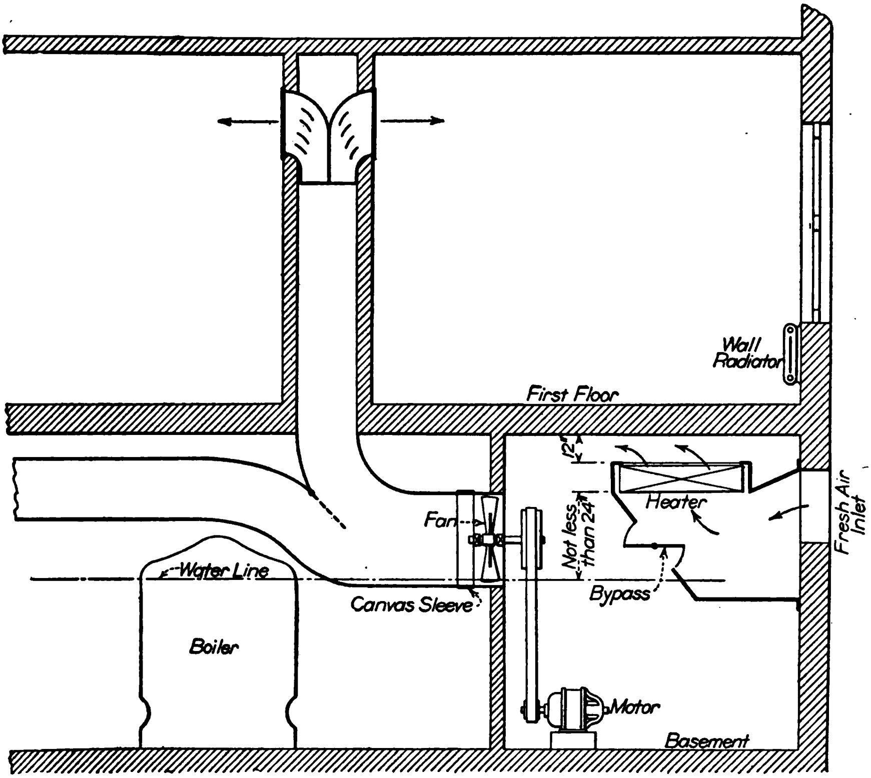


FIG. 49—TYPICAL ARRANGEMENT OF HEATING AND VENTILATING SYSTEM SUITABLE FOR USE IN SMALL SCHOOLS

Tempering Coil and Reheater

Fig. 50 shows a system which provides both a tempering coil and reheater, each consisting of a group one section deep and hung in a horizontal position to provide head room above the water line of the boiler. The reheater is provided with a by-pass and mixing damper, the mixing being accomplished by means of chains and an indicating quadrant from each room. A separate reheater is installed at the base of each flue leading to the various class rooms so the temperature of the entering air can be controlled from each room. This system could be used in cases where it is desired to provide for heating as well as ventilating by introducing the air at a temperature above 70 deg. and installing no direct radiation in the rooms. This would not be advisable, however, unless the conditions were such as to require it. The arrangements shown in Figs. 47 and 48 are applicable only to cases where direct radiation is installed in the rooms.

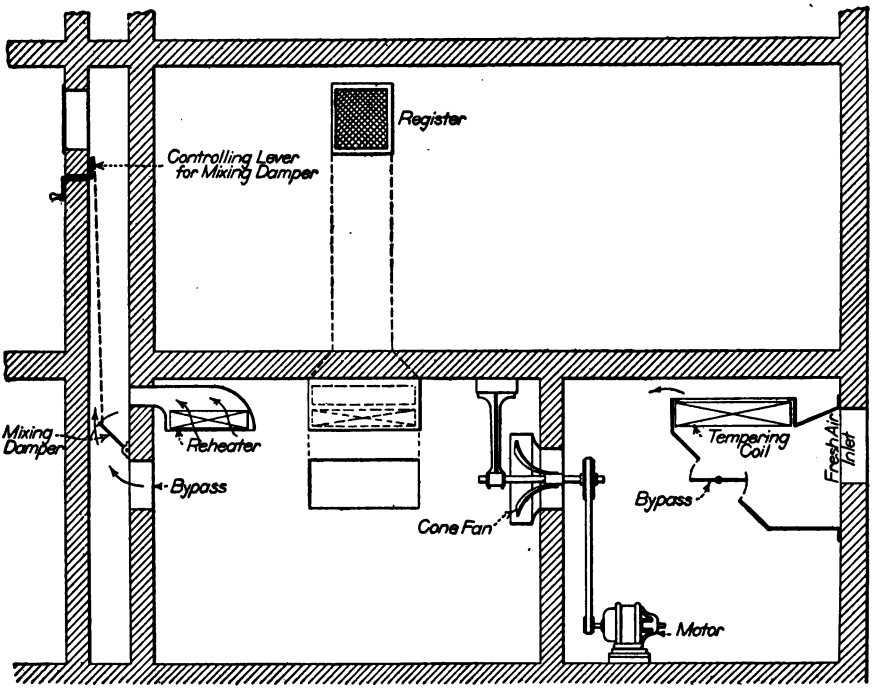


FIG. 50—SYSTEM HAVING BOTH TEMPERING COIL AND REHEATER

CHAPTER XXV

INDIVIDUAL DUCT SYSTEM

IF a system is to be designed to supply air for heating as well as ventilating from a central distributing unit, as may be the case in hotels, etc. (where it is desirable to do away with radiators in some of the rooms), the individual duct system must be used. Each room supplied must have a separate duct leading from the reheater to the supply register in the room. The room may be served by different registers supplied from one duct, but no other duct must supply any of these registers, and no other room may be supplied from this duct.

The heater should be divided into a tempering coil and a reheater previously described. The tempering coil is usually two sections deep and should heat the air to about 40 deg. Fig. 51 shows a typical arrangement for heaters and fan.

The number of sections and the velocity through the reheater must be such as to raise the temperature of the air from 40 deg. to the maximum temperature required in any one of the rooms. It can be seen that the different rooms will not require the same entering temperature of air. Those with a comparatively large amount of exposed wall and glass surface will require air at a higher entering temperature. This variation of temperature in the different ducts is obtained by adjusting the mixing dampers so as to obtain a large proportion of the tempered air and smaller proportion of the hot air when a comparatively low entering temperature is desired. These mixing dampers are controlled automatically from the thermostats located in the room which they supply. The description of these thermostats will be taken up under temperature control.

After the number of sections in the reheater has been determined from the assumed velocity, to determine the number of loops per section, it is not necessary to assume that all the air must pass through the reheater. Assume that the maximum temperature required for any particular room is 130 deg., and that the maximum temperature for some other room is 100 deg. The air which passes through the reheater is therefore at a temperature of 130 deg. and the tempered air through the by-pass below the reheater is at a temperature of 40 deg. Let Q equal the number of cu. ft. of air to be delivered to the second room every hour. If part of this air passes through the reheater, entering the distributing duct at 130 deg., and the remainder, through the by-pass, entering the duct at 40 deg., and as these quantities are in the right proportions, the resulting temperature will be 100 deg., as required. The quantity of heat given up by part of the air cooling from 130

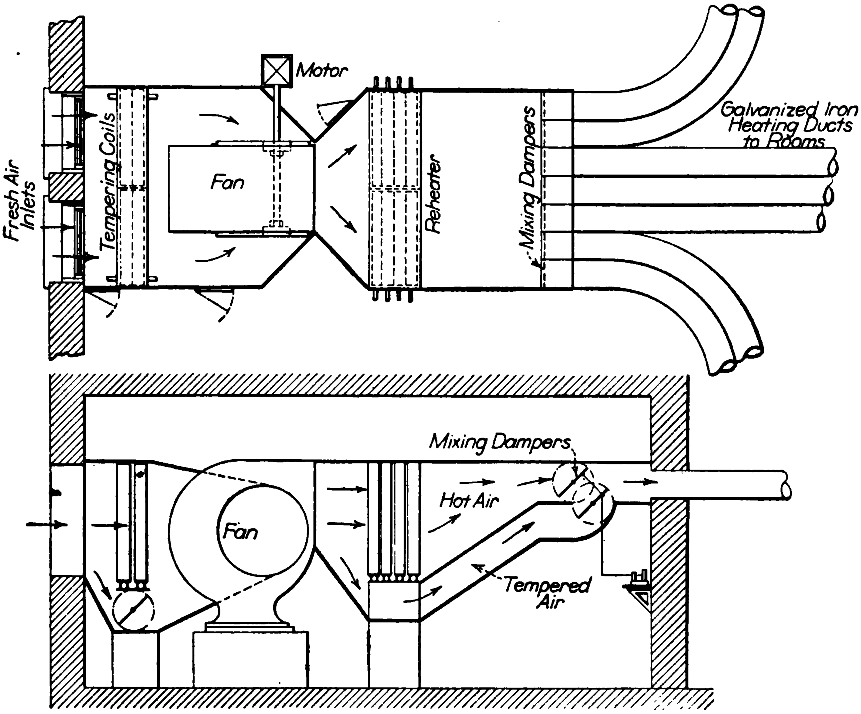


FIG. 51—PLAN AND ELEVATION, SHOWING ARRANGEMENT FOR HEATERS AND FAN

deg. down to 100 deg. must be sufficient to raise the remainder of the air from 40 deg. to 100 deg.

Let A equal the cubic feet of air through the reheater.

Let B equal the cubic feet of air through the by-pass.

The specific heat of air is slightly different at different temperatures and this should be taken into account in determining the quantities accurately, but the variation is so slight that for practical purposes it may be taken as constant at the various points. Referring to formula (1) in Chapter XVII:

$$\frac{Q\ (T_1-T)}{55}=H$$

Substituting the above values we have:

$\frac{B\ (100-40)}{55}=H$, the B. t. u. necessary to raise the air through the by-pass from 40 deg. to 100 deg.

$\frac{A\ (130-100)}{55}=H$ the B. t. u. given up by the air through the reheater cooling from 130 to 100.

These two expressions must therefore be equal.

$$\frac{A\ (130-100)}{55}=\frac{B\ (100-40)}{55}$$

The quantity 55 appears on both sides of the equation, therefore, it can be canceled from each side. Simplifying the expression becomes

$$A\ (130-100)=B\ (100-40)$$

$$\text{or} \qquad A\times 30=B\times 60.$$

$$B=\frac{A\times 30}{60}$$

We also know that the sum of the two quantities A and B must equal the total quantity to the room Q, therefore,

$$A+B=Q$$

Putting the value of B in terms of A from above this becomes

$$A+A\frac{30}{60}=Q$$

$$A\left(1+\frac{30}{60}\right)=Q$$

$$A=\frac{60}{90}\times Q$$

Therefore, to have the resulting temperature of the mixture at 100 deg., $\frac{60}{90}$ or $\frac{2}{3}$ of the air must pass through the reheater and $\frac{1}{3}$ through the by-pass under the above conditions.

To use general terms for any case, let

T_m = the maximum temperature required.

T_b = temperature of air through by-pass.

T desired temperature for any room.

From the above we can write this formula:

$$A\,(T_m - T) = B\,(T - T_b) \qquad (1)$$

$$B = A\left(\frac{T_m - T}{T - T_b}\right) \qquad (2)$$

Substituting the value of B in the formula $A + B = Q$ as above we have

$$A + A\left(\frac{T_m - T}{T - T_b}\right) = Q$$

$$A\left(1 + \frac{T_m - T}{T - T_b}\right) = Q$$

Reducing the quantity in the parenthesis as follows:

$$A\left(\frac{T - T_b}{T - T_b} + \frac{T_m - T}{T - T_b}\right) = Q$$

$$A\left(\frac{T - T_b + T_m - T}{T_w - T_b}\right) = Q$$

$$A\,\frac{T_m - T_b}{T - T_b} = Q \qquad (3)$$

$$A = Q\left(\frac{T - T_b}{T_m - T_b}\right) \qquad (4)$$

To prove this formula, substitute the above values in formula (4)

$$T_m = 130$$
$$T = 100$$
$$T_b = 40$$

$$A = Q\left(\frac{100 - 40}{130 - 40}\right)$$

$$A = Q \times \frac{60}{90}$$

$$A = \frac{2}{3}Q$$

which is the same result as obtained above.

Problem

Assume three rooms, Nos. 1, 2, and 3.

Room No. 1.

$Q = 6{,}000$ cu. ft. per minute.

Temperature of entering air in zero weather must be 130 deg.

Room No. 2.

$Q = 10{,}000$ cu. ft. per minute.

Temperature of entering air in zero weather must be 90 deg.

Room No. 3.

$Q = 2{,}000$ cu. ft. of air per minute.

Temperature of entering air in zero weather must be 100 deg.

Total quantity of air is 18,000 cu. ft. per minute. The tempering coil raises the total quantity of air from zero degrees to 40 deg.

The air passing through the reheater is raised from 40 deg. to 130 deg.

What proportion of the total quantity should pass through the reheater in zero weather and how much through the by-pass?

Solution

All the air for room No. 1. must pass through the reheater —6,000 cu. ft.

Room No. 2.

$$Q = 10{,}000 \text{ cu. ft.}$$
$$T = 90 \text{ deg.}$$
$$T_b = 40 \text{ deg.}$$
$$T_m = 130 \text{ deg.}$$

Substituting in formula (4)

$$A = Q\left(\frac{T - T_b}{T_m - T_b}\right)$$

$$A = 10{,}000\left(\frac{90-40}{130-40}\right)$$

$$A = 10{,}000 \times \frac{50}{90}$$

$A = 5556$ cu. ft. through reheater.

$B = 10{,}000 - 5556 = 4444$ cu. ft. through by-pass.

Proceeding in the same manner with room No. 3.

$$Q = 2000 \text{ cu. ft.}$$
$$T = 100$$
$$T_b = 40$$
$$T_m = 130$$
$$A = 2{,}000\left(\frac{100-40}{130-40}\right)$$
$$A = 2{,}000 \times \frac{60}{90}$$

A = 1333 cu. ft. through reheater.

B = 2000 − 1333 = 667 cu. ft. through by-pass.

Total air through reheater = 6,000 + 5556 + 1333 = 12,889.

Total through by-pass = 4444 + 667 = 5111; 12,889 + 5111 = 18,000.

CHAPTER XXVI

VENTILATING FANS

FANS used for ventilating work may be divided into three general classes: the disk or propeller fan, the cone fan and the centrifugal fan. The disk fan is made in several different forms, the most simple of which is a flat blade, set at a given angle to the plane of revolution as shown in Fig. 52. The propeller fan is a modification of the disk fan, the blades being constructed in the form of a curve instead of being flat. Fig. 53 shows a fan of this type. The cone fan, shown in Fig. 54, operates on the same principle as the centrigufal fan. As the wheel containing the blades revolves, the air contained between the blades tends to move in a straight line, due to inertia. This causes the air to fly off at a tangent to the periphery of the wheel.

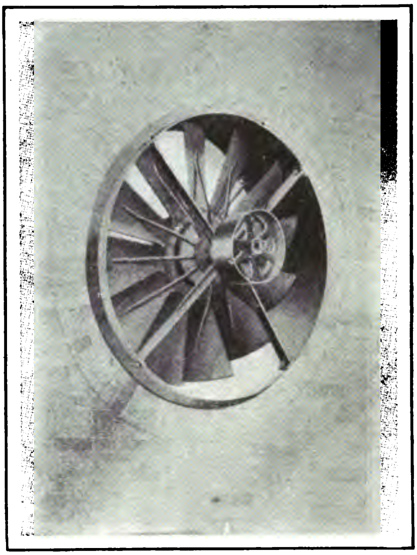


FIG. 52—DISK FAN

When the air between the blades flows out, more air is drawn in from the inlet to take its place and the process is thus continued. There is no housing or casing and the air is discharged equally in all directions in a plane at right angles to the axis of the wheel. One side of the wheel is constructed in the form of a cone from which the fan derives its name. The axis of the cone coincides with the axis of the wheel. This causes all of the air to be drawn in at one side of the fan.

The centrifugal fan, one type of which is shown in Fig. 55, operates on the same principles as the cone fan, with the exception that the wheel is entirely encased with what is known as the housing. The air is collected and flows out in one direction from the outlet. The revolving part is called the wheel. There are two general types of centrifugal fans known as the steel-plate and the multi-blade fan. The only material difference between these two is the construction of the wheels, shown in Figs. 56 and 57 respectively. With the steel-plate fan, the blades are comparatively large and few in number while, with the multi-blade, as the name implies, there are a large number of blades which are much narrower and placed closer than those of the steel-plate fan. They are also constructed on the form of a curve in the direction of rotation.

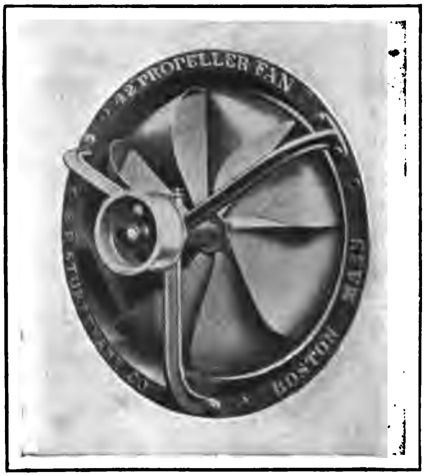


Fig. 53—Propeller Fan

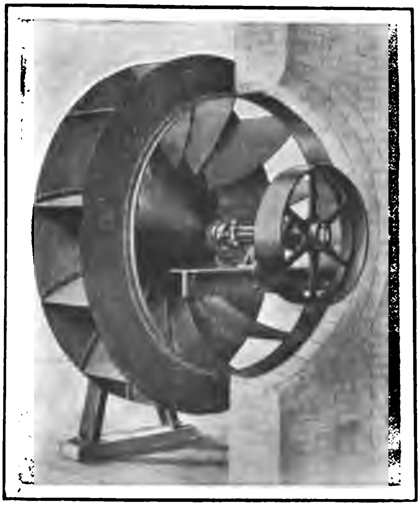


Fig. 54—Cone Fan

The multi-blade fan, though more expensive in first cost than the steel-plate, requires less floor space and head room for the same capacity. It is also slightly more efficient under certain conditions, requiring less power for the same quantity of air delivered. These fans are made with the discharge at any angle so that they can be made to conform to any condition.

Application of Disk Fans

As to the application of the various types given above, the disk and propeller fans are adaptable for handling large quantities of air at very light resistances or what is termed free delivery. They are very efficient and effective when installed in the side of a wall and discharge directly into the atmosphere and the most efficient type of fan for this kind of service.

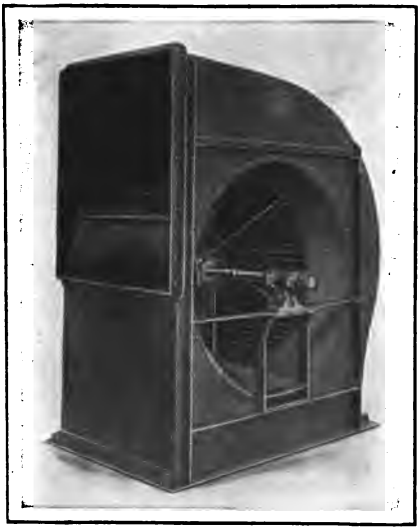


FIG. 55—CENTRIFUGAL FAN

These fans can also be used in connection with the distributing systems if the ducts are short and carefully designed with long easy curves and low velocities. The velocity of flow should never exceed 600 ft. per minute at any point, and should be kept considerably below this if possible. The system should be so designed that the flow of air will always be assisted by the force of gravity and never opposed to it. If a heater is used in connection with the system, the heater should be so designed as to give the proper temperature rise with one stack deep. This requires only a light velocity and low resistance to the passage of air. The proportioning of the heater on this basis has been discussed in a previous chapter.

The cubic feet of air that these fans will handle per minute and the necessary horse-power for driving are given in the manufacturers' catalogues. These ratings, however, are usually given for free delivery and should be decreased at least 50 per cent. and the necessary power for driving should be doubled when used in connection with heaters and ducts, as described above. If the capacities of the fans are given for various resistances, the resistance of the system may be determined as described later and the size of the fan thus selected.

These fans can be used very effectively as exhaust fans in theatres, auditoriums, etc., where the exhaust registers can be so located as to connect directly or through short ducts with the exhausting chamber, the fan exhausting the air from this chamber into the atmosphere.

Cone Fans

The cone fan is capable of working against a much higher resistance than the disk or propeller fan, but the construction of the outlet is such as to make the fan practically useless in connection with a system requiring distributing ducts. The fan is chiefly used in theatre work where it can discharge directly in a large plenum chamber and is particularly adapted to cases where the air must be brought down from the roof or through long ducts and several stacks of heaters before reaching the fan.

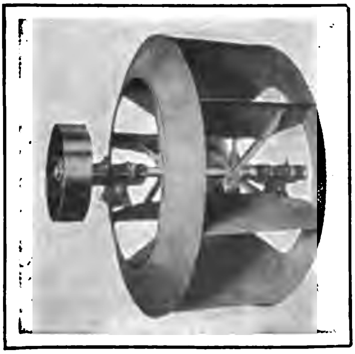


FIG. 56—STEEL PLATE CENTRIFUGAL FAN WHEEL

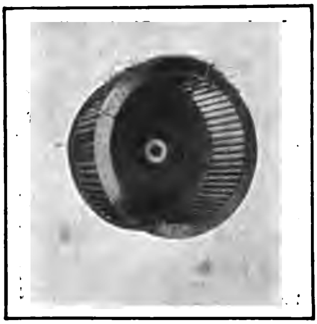


FIG. 57—MULTI-BLADE CENTRIFUGAL FAN WHEEL

The ratings and horse-power for the various sizes of cone fans are given in the catalogues, but the resistance or suction against which the fan is to operate must be known before the size can be selected. The method of determining this will be explained later.

The Centrifugal Fan

Any system requiring three or more stacks of heaters and any considerable amount of duct work through which the air must be delivered to the various rooms, necessitates the use of a centrifugal fan of either the multi-blade or steel-plate type to insure positive results. As a rule, little distinction is made as to which of these two types shall be used, other than that of first cost and space conditions. As pointed out above the multi-blade costs considerably more for the same capacity but requires less floor space and head room. Either one or the other of the above conditions is usually sufficient to determine which of the two types to use. The multi-blade is also slightly more efficient for ordinary ventilating systems where the resistance or pressure against which the fan operates, increases with the increase of the quantity of air. This is due to the fact that the blades are constructed on the form of a segiment of a cylinder, with the curve in the direction of rotation.

Fan Sizes

The various fan manufacturers give complete and reliable tables showing the capacities of the different sizes of fans and the necessary horse-power under various conditions. If these conditions are known, it is a simple matter to select from these tables a fan which will deliver the required amount of air and the necessary size of the engine or the motor to drive the fan. To select the size of the fan needed, two conditions must be known in addition to the cubic feet of air per minute, namely, the speed or revolutions per minute designated by R. P. M. and the pressure at which the fan must operate.

The resistance or friction of the air, moving inside the ducts, causes a pressure to be produced at the outlet of the fan. This pressure is exerted equally in all directions and is called static pressure. In addition to the static pressure, there is also a pressure produced by the velocity or inertia of the air moving along the duct. This pressure is exerted only in the direction

TABLE XXX

No. 6 Single Inlet American Sirocco Fan

Volume Cu. Ft. Per Min.	Outlet Velocity	Tip Speed	¼ in. RPM.	S. P. BHP.	Tip Speed	½ in RPM.	S. P. BHP.	Tip Speed	¾ in. RPM.	S. P. BHP.	Tip Speed	1 in. RPM.	S. P. BHP.	Tip Speed	1½ in. RPM.	S. P. BHP.
12,850	1,800	1,985	210	2.04	2,310	244	2.62	2,600	275	3.28	2,900	307	4.01	3,430	363	5.63
13,550	1,900	2,050	217	2.31	2,370	251	2.97	2,650	280	3.62	2,950	312	4.40	3,450	365	6.01
14,280	2,000	2,116	224	2,58	2,450	259	3.28	2,720	288	4.05	2,980	315	4.74	3,480	369	6.44
15,000	2,100	2,180	231	2.98	2,510	265	3.74	2,770	293	4.40	3,020	320	5.28	3,510	372	6.94
15,700	2,200	2,250	238	3.24	2,570	272	4.13	2,830	300	4.86	3,070	325	5.63	3,550	376	7.48
16,400	2,300		...	...	2,630	278	4,55	2,900	307	5.40	3,120	330	6.17	3,590	380	8.02

TABLE XXXI

No. 7 Single Inlet American Sirocco Fan

Volume Cu. Ft. Per Min.	Outlet Velocity	Tip Speed	¼ in. RPM.	S. P. BHP.	Tip Speed	½ in. RPM.	S. P. BHP.	Tip Speed	¾ in. RPM.	S. P. BHP.	Tip Speed	1 in. RPM.	S. P. BHP.	Tip Speed	1½ in. RPM.	S. P. BHP.
15,520	1,600	1,840	167	2.09	2,200	200	2.87	2,510	228	3.60	2,830	257	4.59	3,400	309	6.53
16,490	1,700	1,900	173	2,40	2,260	205	3.18	2,550	232	4.03	2,850	259	5.06	3,410	310	7.10
17,450	1,800	1,985	180	2.67	2,310	210	3.55	2,600	236	4.46	2,900	264	5.42	3,430	312	7.61
18,430	1,900	2,050	186	3.13	2,370	216	4.02	2,650	241	4.90	2,950	268	5.94	3,450	314	8.15
19,400	2,000	2,116	192	3.50	2,450	223	4.43	2,720	247	5.48	2,980	271	6.42	3,480	316	8.70
20,380	2,100	2,180	198	3.91	2,510	228	5.06	2,770	252	5.95	3,020	275	7.15	3,510	319	9.38

TABLE XXXII

No. 8 Single Inlet American Sirocco Fan

Volume Cu. Ft. Per Min.	Outlet Velocity	Tip Speed	¼ in. RPM.	S. P. BHP.	Tip Speed	½ in. RPM.	S. P BHP.	Tip Speed	¾ in. RPM.	S. P. BHP.	Tip Speed	1 in. RPM.	S. P. BHP.	Tip Speed	1½ in RPM.	S. P. BHP.
19,030	1,500	1,770	141	2.31	2,150	171	3.26	2,480	198	4.34	2,800	223	5.43	.....	....	....
20,300	1,600	1,840	147	2.72	2,200	175	3.74	2,510	200	4.68	2,830	225	5.97	3,400	271	8.48
21,570	1,700	1,900	151	3.13	2,260	180	4.14	2,550	203	5.27	2,850	227	6.58	3,410	272	9.23
22,820	1,800	1,985	158	3,61	2,310	184	4.62	2,600	207	5.76	2,900	231	7.06	3,430	273	9.90
24,100	1,900	2,050	163	4.07	2,370	189	5.23	2,650	211	6.39	2,950	235	7.73	3,450	275	10.6
25,380	2,000	2,116	168	4.55	3,450	195	5.77	2,720	217	7.13	2,980	237	8.35	3,480	277	11.3

of flow and is called velocity pressure. The sum of these two pressures is designated as total pressure. These pressures are usually measured in inches of water which means the number of inches high that a column of water would be sustained by the pressure. In rating the capacity of a fan the static pressure only, is usually given. Tables 30, 31 and 32 give the capacities of three different sizes of American Sirroco fans, manufactured by the American Blower Co., under different speeds and static pressures from ¼ in. to 1½ in.

The first column gives the cubic feet per minute delivered by the fan. The second column is the outlet velocity, that is, the velocity of the air in feet per minute leaving the outlet of the fan for the corresponding quantities given in the first column. The columns marked tip speed are the speeds of the tip of the blades in feet per minute for the corresponding revolutions per minute given in the adjacent columns. The columns marked B. H. P. are the brake horse-powers necessary to deliver the corresponding quantities of air at the various pressures.

From the accompanying tables it can be seen that it is absolutely essential to know approximately what the static pressure of a system will be in order to determine the proper size of fan, since a slight variation in the pressure causes a considerable variation both in the cubic feet of air that a certain fan will handle and the horse-power to drive the fan. The static pressure is also the most difficult part of the computation to estimate with any degree of accuracy as there are so many factors which enter in to influence it. It may be computed from the plans by means of formulas but, when the duct work is installed, certain structural conditions may enter in, causing various modifications from the plans which will alter considerably the computed results. It therefore requires considerable experience and judgment in addition to mathematical calculations. The following general rules for determining the static pressure may be used with a fair degree of safety but should be modified to suit conditions and also checked by means of formulas and rules given later.

For small buildings and schools having comparatively short runs with ducts designed with easy curves, the velocities as outlined previously and the heater composed of three stacks use from ⅝-in. to ¾-in. static pressure, depending upon length of duct work. If four or five sections are used in the heater and mixing dampers with individual ducts to rooms, use from ¾-in. to ⅞-in. static pressure. With large complicated systems such as are encountered in hotel work, etc., where the ducts are long with many turns, use from 1 in. to 1¼-in. static pressure.

If an air washer is used, the above figures should be increased about ¼-in. in each case.

The speed at which a fan should operate, depends entirely upon the importance of noiseless operation and first cost as compared to economy. In general the larger the fan, the less power it requires to deliver the same quantity of air against the same pressure. This fact will be observed from examination of the above tables. For schools and similar installations where the noise must be kept to a minimum, the tip speed of the wheel should not exceed 2,800 ft. per minute or the outlet velocity 2,200 ft. per minute.

From the more complete tables given by manufacturers, it will be found that four or five different sizes of fans may be selected which will deliver the required amount of air against the estimated static pressure but the smaller the fan, the higher will be the tip speed and the greater the power. The proper size, however, should be determined from a careful study of the conditions as outlined above.

Assume for illustration, that it is required to select a fan for a school ventilating system where the quantity of air is 15,500 cu. ft. per minute and the estimated static pressure is ¾-in. Referring to the table No. 30 we find that a No. 6 fan will deliver 15,700 cu. ft. per minute with an outlet velocity of 2,200 ft., a tip speed of 2,830 ft. per minute the required horse-power being 4.86. From table 31 a No. 7 fan will deliver 15,520 cu. ft. per minute with an outlet velocity of 1,600 ft., a tip speed of 2,550 ft. per minute, and the required horse-power is 3.6. The latter fan would be better adapted for the conditions.

CHAPTER XXVII

ESTIMATING STATIC PRESSURES

IN the previous chapter, several rules were given for determining roughly the static pressure in various kinds of ventilating systems. It is advisable however, to check up these figures mathematically, if possible, and the following formulas and rules may be used to determine accurately the static pressure in any system. The static pressure of any point in a duct through which air is flowing, is caused by the friction of this air moving in contact with the sides of the duct between the point at which the pressure is measured and the outlet of the duct. The empirical formula most generally used for determining this friction is as follows:

$$P = \frac{K\,S\,V^2}{A} \qquad (1)$$

in which the symbols are:

P = loss of pressure or friction in inches of water gauge.
K = constant depending upon the construction of the duct.
S = total internal rubbing surface of the duct in square feet.
V = velocity of the air feet per second.
A = area, in square inches.

The value of K is usually taken as 0.0002 for galvanized iron ducts and 0.00028 for brick or concrete ducts.

This formula is only for straight ducts. Where bends occur, these should be expressed in equivalent length of straight pipe. These are usually given in terms of the number of diameters of the pipe. Table No. 33 gives the number of diameters or widths of straight pipe to allow for 90-deg. bends when the radius of the throat is expressed in terms of the diameter of the pipe.

To illustrate the application of the above, assume a duct 36 in. in diameter with a 90-deg. bend, the radius of the throat of which is 27 in. The ratio of the throat to the diameter is then 27 to 36 or ¾.

From table XXXIII, the equivalent friction effect is 16 diameters. The duct is 36 in. in diameter or 3 ft., therefore, the elbow has the same loss in friction as 16×3 = 48 ft. of straight duct.

TABLE XXXIII

Radius of Throat to Diameter or Width of Pipe	Equivalent Numbers of Diameters of Straight Pipe
0 (square term)	100
¼	67.0
½	30.0
¾	16.0
1	10.0
1¼	7.5
1½	6
2 and above	5

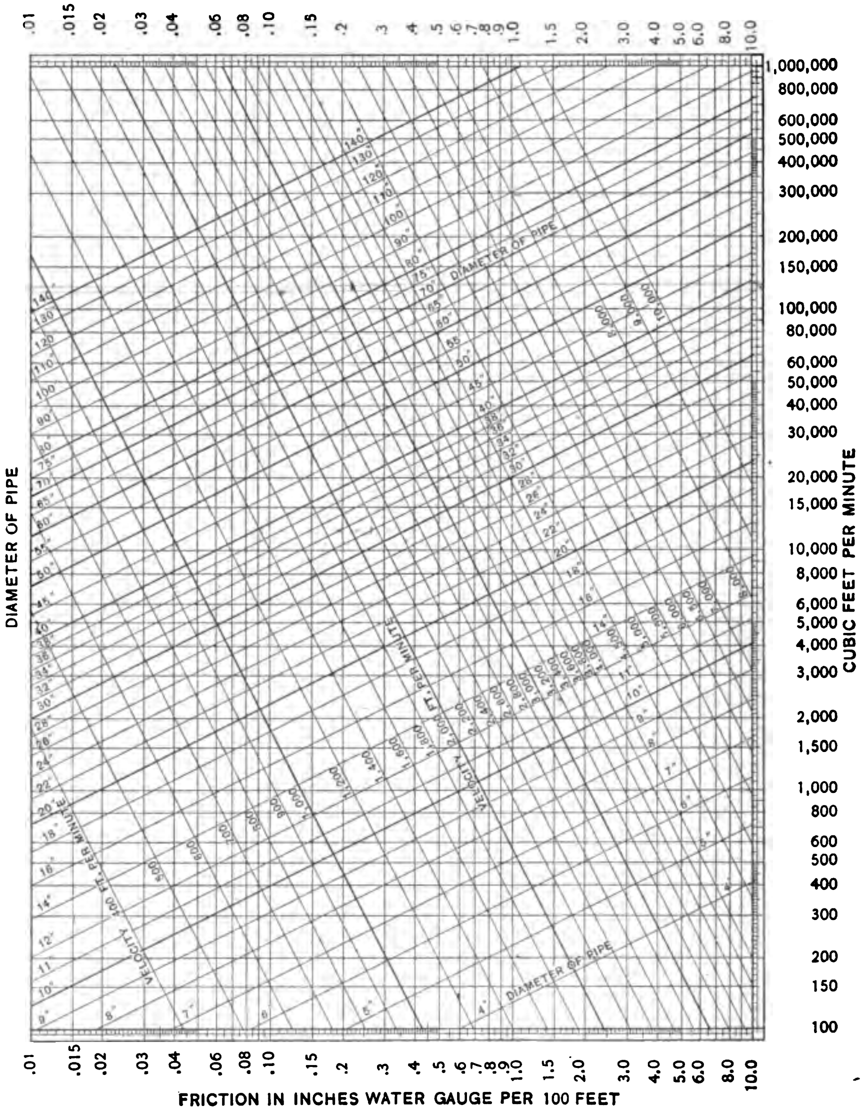
For rectangular ducts the method of procedure is the same, taking the ratio of the throat to the width of the duct in the plane in which it bends. If the bend is more or less than 90 deg. the result is estimated first on a basis of 90 deg. and then multiplied by the ratio of the angle to 90 deg. If in the above assumed case, the elbow had been 45 deg. instead of 90 deg., the result would have been one half of 48 or 24 ft. of straight duct.

Chart No. 1 gives the loss in friction in inches of water for round ducts from 4 in. in diameter up to 140 in. in diameter and for velocities from 400 ft. per minute up to 10,000 ft. per minute.

The diagonal lines running up to the right, indicate the diameter of the ducts in inches. The diagonal lines running up to the left represent the different velocities in feet per minute. The horizontal lines represent quantities in cubic feet per minute which are indicated on the right side of the chart. The vertical lines represent the friction loss in inches of water for 100 ft. of straight duct which are given at the top and bottom of the chart. The values are determined from formula (1) given above.

To illustrate the application of the chart, assume a duct 200 ft. long and 40 in. in diameter in which there is one 90-deg. bend with a throat radius of 20 in. and the velocity in the duct

Chart No. 1


FRICTION IN INCHES WATER GAUGE PER 100 FEET
.01
.015
.02
.03
.04
.06
.08
.10
.15
.2
.3
.4
.5
.6
.7
.8
.9
1.0
1.5
2.0
3.0
4.0
5.0
6.0
8.0
10.0
CUBIC FEET PER MINUTE
1,000,000
800,000
600,000
500,000
400,000
300,000
200,000
150,000
100,000
80,000
60,000
50,000
40,000
30,000
20,000
15,000
10,000
8,000
6,000
5,000
4,000
3,000
2,000
1,500
1,000
800
600
500
400
300
200
150
100
DIAMETER OF PIPE
140"
130"
120"
110"
100"
90"
80"
75"
70"
65"
60"
55"
50"
45"
40"
38"
36"
34"
32"
30"
28"
26"
24"
22"
20"
18"
16"
14"
12"
11"
10"
9"
8"
7"
6"
5"
4"
VELOCITY 400 FT. PER MINUTE
500
600
700
800
900
1,000
1,200
1,400
1,600
1,800 FT. PER MINUTE
VELOCITY 2,000
2,200
2,400
2,600
2,800
3,000
3,200
3,400
3,600
3,800
4,000
4,500
5,000
5,500
6,000
6,500
7,000
7,500
8,000
9,000
10,000

as 1400 ft. per minute. To find the equivalent length of straight duct in the bend, $\frac{20}{40} = \frac{1}{2}$ which is equivalent to 30 diameters from the table, $\frac{40}{12} \times 30 = 100$ ft. Total length is, therefore, $200 + 100 = 300$ ft. Referring to Chart No. 1, find the diagonal line marked 40 in. Follow this up to the right until it intersects the 1,400 ft. velocity line. At the point of intersection of these two lines, follow a vertical line down the bottom of the chart. This point is found to lie midway between 0.08 and 0.10. Therefore, the friction per 100 ft. is 0.09. The total length of the duct is 300 ft. and the total friction loss would be $\frac{300}{100} \times 0.09 = 0.27$ in. The chart can also be used to determine the cubic feet of air flowing through the duct per minute in this case. At the point of intersection of the 40-in. diameter line with 1,400 ft. velocity line, follow horizontally to the right. The quantity is found to be about 12,500 cu. ft. per minute.

If the system consists of several different branch ducts, leading off from a main duct, as is often the case, the longest run from the source should be the one taken to determine the total friction. The first step is to go over the entire length of duct selected, indicating the velocities in each of the different sized sections and measure their lengths. Then determine the equivalent lengths of each bend as shown above. The next step is to determine the loss in friction for each section from the chart. The total loss in friction will then be equal to the sum of the frictions in the various sections.

Chart No. 1 may be still further used in the design on the ventilating ducts for the determining of areas and sizes of the ducts. The method previously described was to assume the velocities at the various points and from these assumed velocities, determine the sizes. This method is only approximate and necessities resorting to the use of deflectors and volume dampers to obtain the proper distribution of the air to the various points. This method is sufficiently accurate for ordinary size systems where the runs are short, but for large systems it introduces unnecessary resistance because of the number of deflectors and dampers which often have to be set at sharp angles to produce the proper flow. Where the system is large with long runs, it is advisable to design the duct work on what is known as a drop in pressure basis, that is, the drop in pressure per unit of length will be the same in all points of the main duct, regardless of the quantity of air flowing. Furthermore, the total drop in pressure in any branch duct, from the point where it leaves the main duct to the last outlet, should be the same as the total drop in pressure in the main duct from this point to the end of the main duct. To illustrate this latter point, assume a duct system as shown in Fig. 58. The equivalent lengths of the various sections are given on the plan. These lengths include the proper allowance for the various bends. The quantity of air to be delivered is also indicated at the outlets. The main or longest duct is the section A-E.

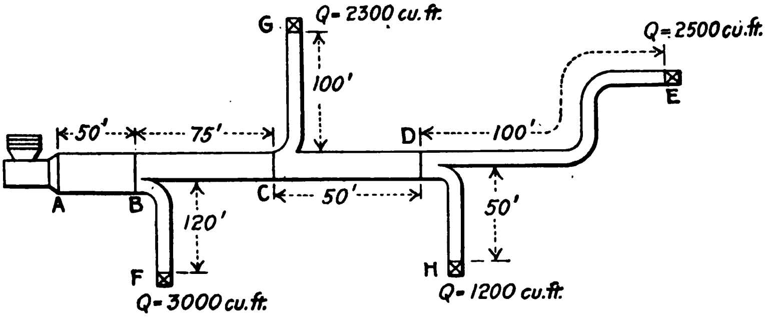


FIG. 58

In proportioning the branch duct D. H. the drop in pressure between D and H should be the same as between D and E. The drop in C-G should be the same as the drop between C and E, etc. If this is done, the size of the outlet for the branch of C (or any other part) and the main duct can be proportioned on the same velocity basis, whatever the velocity may be and the proper quantity of air will flow to the branch. If the drop in pressure between the points C and the outlet G were less than the drop in pressure between C and E for the quantities assumed, then more air would flow out of the branch and less through the main duct, increasing the resistance in the branch until the conditions were equalized. Proper flow of air must then be established by setting the deflectors at C at an angle to the flow of air, thus inducing additional resistance in the duct and consequently increasing the power to deliver the required quantity of air.

In order to obtain some basis on which to proceed with the proportioning of the main duct, the friction loss or drop per 100 ft. of length must first be established. This could be done in two ways. By assuming a total drop in pressure for the entire system, this divided by the total length on the main duct in feet and multiplied by 100 would give the drop per 100 ft. The maximum velocity at some point might be assumed and the drop per 100 ft. determined from the chart.

In the above case, assume a limiting velocity of 1,200 ft. per minute in the main duct from the fan or the section A. B. The total quantity of air through this section is the sum of the quantities at F. G. H. and E. or 9,000 cu. ft. per minute.

Referring to the chart at the right hand side "Cubic Feet per Minute" find the quantity 9,000. Follow this horizontally to the left until it intersects the 1,200 ft. velocity line. This point is midway between the 36-in. diameter line and the 38-in. diameter, therefore, the diameter of the section A-B should be 37 in. From this point follow down to the bottom of the chart and the friction is found to be about 0.07 in. per 100 ft. This friction drop can then be taken as the basis from the design of the main duct. Starting now from the end or outlet E and working back, Q for the section D. E. is 2,500 cu. ft. per minute. Find 2,500 on the right hand side of the chart. Follow this to the left until it intersects the 0.07-in. vertical friction line which point is found to lie on the 22-in. diameter line. The section D. E. should therefore be 22 in. in diameter. The velocity in this section can also be determined at the same time and is found to be about 920 ft. per minute. The total length is 100 ft., therefore, the total drop is 0.07 in. The quantity of air in the section C. D. is 2,500+1,200=3,700. Finding the intersection of the 3,700-ft. line with 0.07-in. friction line, the diameter of this section is found to be 26 in. The drop from C to D is $\frac{50}{100}\times 0.07 = 0.035$ in. The total drop from C to E is 0.035+0.07 = 0.105 in. The velocity in this section is 1,000 ft. per minute. The section B. C. carrying 3,700+2,300 = 6,000 cu. in. per minute is found in the same manner to be 32 in. in diameter and a velocity of 1,100 ft. per minute.

The drop from B to C will be $\frac{75}{100} \times 0.07 = 0.0525$ in. The total drop from C to E is 0.105+0.0525 = 0.1575 in. The branch duct from G to H must have the same drop as D. E. or a total of 0.07. The length D H is 50 ft. The chart is made for runs of 100 ft., therefore the friction per 100 ft. must be determined for this section which will be $\frac{100}{50} \times 0.07 = 0.14$ in.

The quantity, 1,200 cu. ft. per minute, must be carried across the intersection of the .14-in. line. This requires a 14¼ in. duct and the velocity is 1,060 ft. per minute. The total drop in the section C G should be the same as the total drop from C to E or 0.105 in. This run being 100 ft. the size is determined directly from the 0.105 friction lines and is found to be 20 in. in diameter and the velocity 1,080 ft. per minute. In the same manner, the drop in the section B. F. must be 0.1575 in. or $\frac{100}{120} \times 0.1575 = 0.13$ in. per 100 ft. The diameter is slightly over 21 in., use 22 in. and velocity 1,200 ft. per minute.

The friction drop in the section A. B. is $\frac{50}{100} \times 0.07 = 0.035$ and the total drop in the system .1575+0.035 = 0.1925 in. from the point to the outlets.

Rectangular Ducts

Chart No. 1 may also be used in connection with rectangular ducts as well as round ducts by first determining the diameter of a round duct which has the same friction loss as the rectangular duct, then using this diameter in connection with a chart. It can readily be seen that a wide flat duct would have more loss in friction per 100 ft. than a square duct of the same area because of the increased rubbing surface and therefore the equivalent diameter of a round duct would be smaller. Or, if the wide flat duct is to carry the same quantity of air with the same drop in pressure, the area must be larger and the velocity less. Referring to the formula (1) for friction loss $P=\frac{K\,S\,V^2}{A}$.

Let a round duct, the diameter of which is D, be assumed, and a rectangular duct, the sides of which are a and b; then, substituting the value of D in the above fomula, S, the rubbing surface, becomes equal to the circumference of the duct times the length. If 100 ft. is assumed for the length, then

$$S=\Pi \times D \times 100$$

$$A=\frac{\Pi\,D^2}{4}$$

Therefore,

$$P=\frac{K\times\Pi\times D\times 100\times 4\times V^2}{\Pi\times D^2}=\frac{K\times 400\times V^2}{D} \qquad (1)$$

For the rectangular duct—

$$S=(2a+2b)\times 100$$

$$A=a\times b$$

Therefore,

$$P=\frac{K\,(a+b)\,200\times V^2}{a\times b} \qquad (2)$$

Now assume a round duct and a rectangular duct each 100 ft. long having the same velocity, that will be the relation between the two with equal drop in pressure? If the drop in pressure is to be equal, then the two expressions above for P will be equal to each other.

Equating these have:

$$\frac{K\times 400\times V^2}{D}=\frac{K\,(a+b)\,\times 200\times V^2}{a\times b}$$

If the ducts are constructed of the same material, then K is on each side of the equation, and may be cancelled. The velocities in the two ducts are to be equal therefore, the V^2 on each side may be dropped. The expression then becomes:

$$\frac{400}{D}=\frac{200\,(a+b)}{a\times b}$$

$$\frac{1}{D}=\frac{a+b}{2(a\times b)}$$

Inverting the equation,

$$D = \frac{2\,a\,b}{a + b} \tag{3}$$

which gives the relation between round and rectangular ducts where the velocities are equal. Assume as duct 10 x 30 in.,

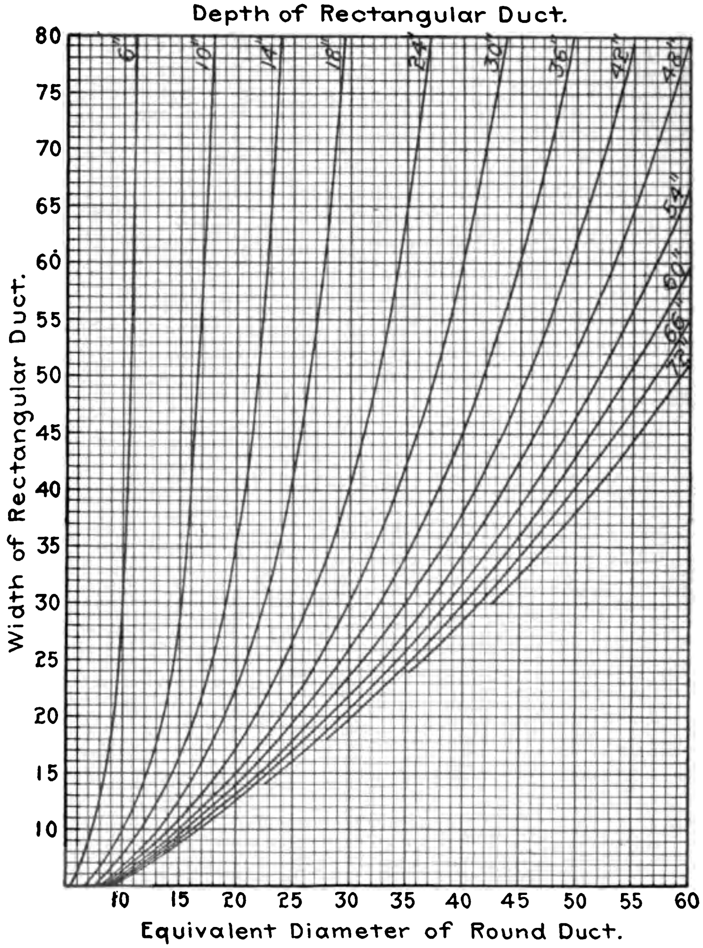


CHART NO. 2—SHOWING RELATION BETWEEN RECTANGULAR DUCTS AND ROUND DUCTS FOR EQUAL FRICTION LOSS WHEN VELOCITIES ARE EQUAL then the values of a and b are 10 and 30 respectively. Therefore $D=\frac{2\times10\times30}{10+30}=15$ in. A 10 x 30-in. duct will therefore have the same friction loss per round 100 ft. as a round duct 15 in. in diameter when the velocities are equal. Assume a velocity of 1200 ft. per minute in the 10 x 30-in. duct, then the friction per 100 ft. may be found by referring to the chart No. 1 and finding the friction in a 15-in. duct with the above velocity.

In chart No. 2, the values of the above formula are plotted in curves for values of a and b from 6 in. up to 80 in. and the values of the diameter may be determined directly from this chart without the necessity of solving the formula.

At the left is given one side of the rectangular duct in inches. The curved lines on the chart represent the opposite side of the duct. At the bottom are the corresponding diameters of round ducts. To illustrate the use of the chart, assume a duct 10 x 34 in., find the point 34 at the left. Follow horizontally to the right until this line intersects the 10-in. curved line. From this point of intersection, follow down to the bottom of the chart and the corresponding diameter is found to be 15.6 in. The friction loss in the 10 x 34-in. duct can therefore be found from chart No. 1 by using a duct 15.6 in. in diameter where the velocity is known. It must be noted, however, that this chart or the formula can be used only when the velocities in the two ducts are equal and the quantity of air flowing through the ducts or the carrying capacities of the two are not necessarily equal.

Relative Carrying Capacities of the Rectangular and Round Duct

To determine the relative carrying capacity of rectangular ducts and round ducts or the size of a rectangular duct which will carry the same quantity of air as a round duct with the same friction loss we require a different method of procedure and a slightly different chart than the one given above. Referring to formula (1) for friction loss $P=\frac{K\,S\,V^2}{A}$, the quantity (V^2) may be replaced by $\left(\frac{Q}{A}\right)^2$ where Q = cubic feet per second and A = area in square inches. The equation then becomes

$$P=\frac{K\,S\,Q^2}{A^3} \qquad (4)$$

which expresses the friction loss in a duct in terms of the quantity of air per second and the area of the duct in square inches. (A) for the round duct is as before $\frac{\Pi\,D^2}{4}$ and $A^3=\frac{\Pi^3\,D^6}{64}$.

$$S=\Pi\times D\times 100$$

Substituting these values

$$P=\frac{K\times\Pi\times D\times 100\times Q^2\times 64}{\Pi^3\times D^6}$$

$$P=\frac{K\times 6400\times Q^2}{\Pi^2\times D^5} \qquad (5)$$

which gives the friction loss in a round duct in terms of the cubic feet of air per second and the diameter in inches.

Substituting the value of A for a rectangular duct in eouation (4)

$$A^3=a^3\times b^3$$

$$S=2(a+b)\times 100$$

$$P=\frac{K\times 2(a+b)\times 100\times Q^2}{a^3\times b^3} \qquad 6)$$

which gives the friction loss in rectangular ducts in terms of cubic feet of air per second and the sides of the duct in inches. As it is desired to establish relationship between round and rectangular ducts of the same friction loss per 100 ft., the two expressions for (P), (5) (6) may be equated as before, therefore,

$$\frac{K\times 6400\times Q^2}{\Pi^2\,D^5}=\frac{K\times 200\ (a+b)\ \times Q^2}{a^3\times b^3}$$

The value K is to be the same in each, therefore, it may be dropped from each side of the equation. The ducts are to have the same carrying capacity or (Q) must be equal in each expression, therefore, it may be dropped from each side and the expression becomes

$$\frac{32}{\Pi^2\,D^5}=\frac{a+b}{a^3\times b^3}$$

Inverting the equation and solving for D we have

$$(D)^5=\frac{32\ (a^3b^3)}{\Pi^2\ (a+b)}$$

$$D = 2\sqrt[5]{\frac{a^3 b^3}{\Pi^2 (a+b)}} \qquad (7)$$

This formula is too difficult and long to solve for each specific case so the values have been plotted in the form of curves as before and shown in friction chart No. 3. The values are

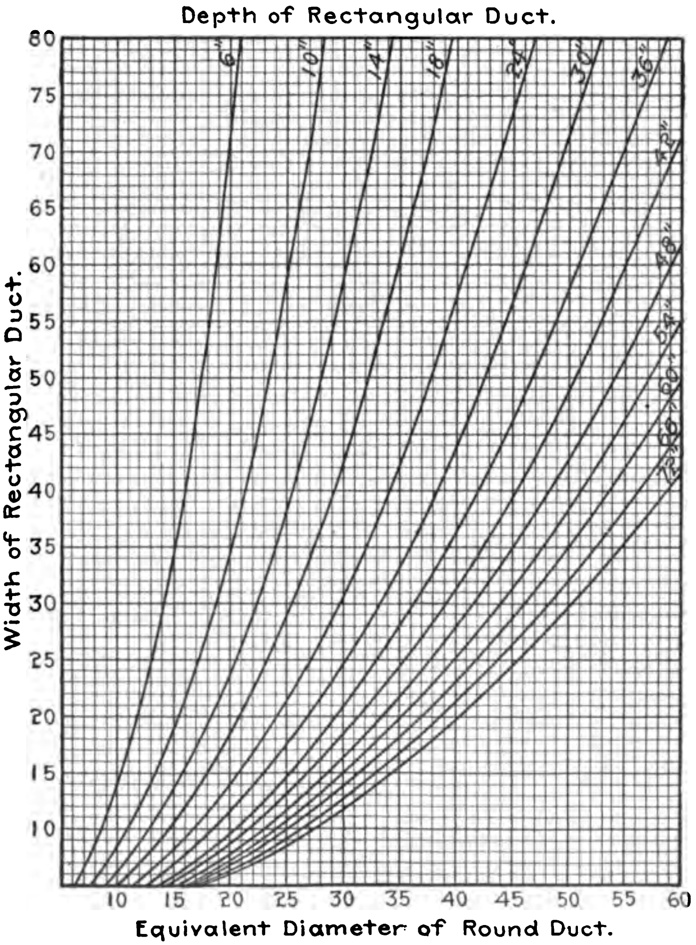


CHART NO. 3—SHOWING RELATION BETWEEN RECTANGULAR DUCTS AND ROUND DUCTS BETWEEN EQUAL FRICTION LOSS WHEN QUANTITIES ARE EQUAL represented in the same manner as in chart No. 2. One side of the rectangular duct is given at the left of the chart, the other side is represented by the curved lines and the corresponding diameter is found at the bottom. This chart, used in connection with chart No. 1, may be used in the design of a complete system with rectangular ducts in the same manner as for round ducts. The method of procedure would be, first, to design the system for round ducts as demonstrated above, then from chart No. 3 determine the corresponding size of rectangular duct to use in each case, by fixing either the width or the depth as the conditions may require.

Assume that the system shown in Fig. 55 is to be designed for rectangular ducts instead of round ducts and that all ducts are to be made 14 in. deep. The section D E was 22 in. in diameter. On chart No. 3 find the quantity 22 at the bottom. Follow this line up vertically until it intersects the 14-in. curved line. From this point of intersection, follow a line horizontally to the left, it is found to be on the 29 in. line. Therefore, this section should be 14 x 29 in. It will then carry the same quantity of air 2,500 cu. ft. per minute with the same friction loss as before namely, 0.07 in. It should be noted that the velocity will be slightly less than with the round duct.

The section C D was 26 in. in diameter. In the same manner this is found to require a duct 14 x 42 in. and the friction drop is as before, 0.035 in. for the section.

Problem

Determine the dimensions of the remaining part of the system and also the velocities in each.

Friction Through Heaters

If the static pressure against which a fan is to operate is to be determined mathematically, the friction through the heater must also be estimated as well as the friction of the ducts. As a rule the largest part of the friction is caused by the air passing through the heaters. The following tables Nos. 33 and 34 determined from tests, give the friction loss through both pipe coil heaters and vento heaters for different velocities and various depths of sections. The friction loss for any standard condition can be selected from these tables.

TABLE XXXIV

Pipe Coils

FRICTION IN INCHES WATER GAUGE

Velocity Air in Feet per Minute	Number of Sections Deep 1	2	3	4	5	6
800	.05	.09	.13	.18	.22	.26
1000	.08	.15	.21	.28	.35	.41
1200	.12	.21	.30	.40	.50	.59
1800	.26	.47	.68	.90	1.11	1.32
2000	.32	.58	.85	1.11	1.37	1.63
2400	.39	.70	1.02	1.34	1.66	1.97

TABLE XXXV

Regular Section Vento

FRICTION IN INCHES WATER GAUGE

Velocity Air in Feet per Minute	Number of Stacks Deep 1	2	3	4	5	6	7	8
600	.017	.030	.042	.055	.067	.080	.093	.105
800	.030	.052	.074	.097	.120	.142	.165	.187
1000	.047	.082	.117	.152	.187	.223	.258	.293
1200	.067	.118	.169	.219	.269	.320	.371	422
1400	.092	.161	.230	.299	.369	.437	.506	.574
1600	.120	.210	.300	.390	.480	.570	.660	.750
1800	.152	.266	.380	.494	.608	.722	.835	.949

From the above tables, the loss in friction through the heating stacks may be estimated for any average condition. The sum of this loss, plus the loss in the duct, will give the total loss in the two. To this, about 15 per cent. to 25 per cent. should be added for losses at the entrance of the fresh-air intake, through the register, for changes in velocity, etc. If this method is followed the actual static pressure against which the fan is to operate should be determined with a fair degree of accuracy.

The location of the heaters with respect to the fan, makes no difference with the total static pressure against which the fan must operate. The total pressure in every case is the difference between the pressure at the inlet of the fan and the outlet. If the inlet to the fan is open to the atmosphere or as is often the case, if the fan draws the air directly from an open room, and blows through the heater and then through the distributing ducts, the pressure at the inlet of the fan is atmospheric or zero pressure. The total static pressure of this system would be a certain number of inches water gauge above atmosphere. This could actually be measured by taking readings with the proper pressure gauge directly at the outlet of the fan. If the fan were placed between the heater and the duct, drawing the air through the heater and then forcing it through the duct, the pressure at the inlet of the fan would be below atmosphere and a certain number of inches of vacuum would be represented by the friction through the stacks. The pressure at the outlet of the fan would be a certain number of inches pressure above atmosphere represented by the friction in the ducts. The actual pressure at the outlet of the fan above atmosphere is not as much in this case as in the former but the total pressure which would be the sum of the loss through the heater plus the loss through the ducts would be the same as before.

This is illustrated by Fig. 59 and Fig. 60. Fig. 59 represents the first case in which the inlet to the fan is not restricted and the air is blown through the heater and then through the fan. The loss in friction through the ducts is assumed to be 0.25 in. water gauge and through the heater 0.5 in. water gauge, making a total of 0.75 n. as shown. At point A, the outlet of the fan, the pressure would read by measurement 0.75 in. At point B between the heater and the duct, the pressure would read 0.25 in. above atmosphere. Directly at the outlet of the duct the static pressure is, of course, zero. The total pressure will be the sum of the 0.25 in. and 0.50 in. or 0.75 in. Fig. 60 represents the second case with the fan placed between the

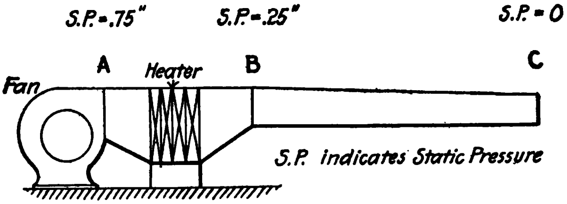


FIG. 59

heater and the duct so that the air must be drawn through the heater. With this case, the pressure at the point A between the fan and the heater will be 0.5 in. below atmosphere or a vacuum of 0.5 in. At point B, the outlet of the fan, the pressure will be 0.25 in. above atmosphere as before. The difference between the absolute pressure at the inlet of the fan and the outlet of the fan is the same as before, 0.75 in. which is the total pressure against which the fan must operate.

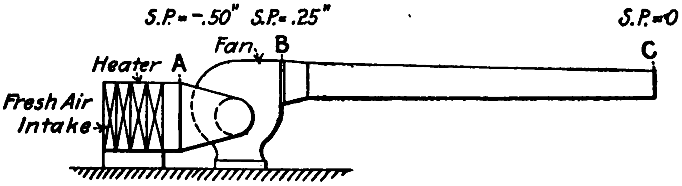


FIG. 60

If the total pressure were to be measured in this case, one leg of the water gauge should be connected at the point B and the other leg at the point A. The gauge would then read total pressure of this system.

CHAPTER XXVIII

FAN MOTORS

ONE of the most important features of a ventilating system with motor driven apparatus is the motor, and the success of the system depends to a large extent upon the selection of the proper type of motor and the controlling apparatus. If direct current is available, the best arrangement is to have the motors connected direct to the fan. This is particularly true in school installations where every precaution must be taken to eliminate noise. This requires, of course, a slow speed motor, since the best types of slow speed motors operate with practically no noise, while with the high speed, there is always a magnetic hum of more or less intensity, which is objectionable. The cause of noise is as liable to be in the faulty operation of the motor as from vibration in the fan. In addition to this, there is also the noise of the belt or driving apparatus.

The one objection to a slow speed motor is the first cost. The slower the speed, the larger and heavier the machine must be for the same power, and consequently the first cost for the same horse-power is inversely proportional to the speed. In factory work and similar installations, where the question of noise is of little importance, belt or chain driven apparatus with high speed motors may be used, when it is desirable to keep down the cost of the installation.

The actual brake horse-power necessary for fan service has been discussed under subject of Fans and may be selected with sufficient accuracy directly from the tables of fan capacities given in the various catalogues. In selecting the size of a motor, however, for any particular installation, an allowance of 10 per cent. to 15 per cent. increase over the actual estimated power should be made for reserve capacity. An additional allowance of about 15 per cent. should be made if belt drive is to be used.

Direct Current Motors

If direct current is used, the motor should be the shunt wound, variable speed type, which makes the system much more flexible than if a constant speed motor is used. This arrangement is again very desirable in school systems as it provides a means of saving both in current consumption and coal by operating the fan at a reduced speed when the school is not in session. It may also happen that, after a system is installed and is operated at the rated speed, more or less air may be delivered than is required, due to more or less resistance in the duct system than was originally estimated. In either case, it is desirable and necessary to change the speed of the fan until the proper conditions are established. With a variable speed motor and controlling apparatus, this may be accomplished by simply setting the controller at various points until the speed which gives the proper quantity of air is found.

Variable Speed Direct Current Motors

In order to change the speed of a direct current motor, either one of two things must be done, the quantity of current flowing through the field must be changed, or the quantity flowing through the armature must be changed. As would naturally be assumed, if the quantity of current flowing through the armature were decreased, the speed of the motor would decrease. With the current flowing through the field, however, the reverse of this is true; that is, if the quantity of current flowing through the field is decreased, the speed increases, or if the current increases, the speed decreases. This is true because the field current produces magnetic lines of force between the two poles in which the armature revolves. The armature revolving in these lines of force, establishes what is known as a counter electro motive force. This produces a current which tends to flow in opposition to the current flowing through the armature. The higher the speed of the armature, the stronger this opposing force becomes. When a motor is started, the speed increases until this counter electro motive force is sufficient to balance the force of the current flowing through the armature, therefore, if the counter electro motive force is decreased by decreasing the current through the field, the motor will speed up until the conditions are again balanced.

In a shunt wound motor, part of the current to the motor flows through the field and the remainder flows through the armature. From the above it can be seen that the speed of the motor may be changed by varying the quantity of current through the armature, which is called "Armature control;" or it may be changed by varying the field current which is

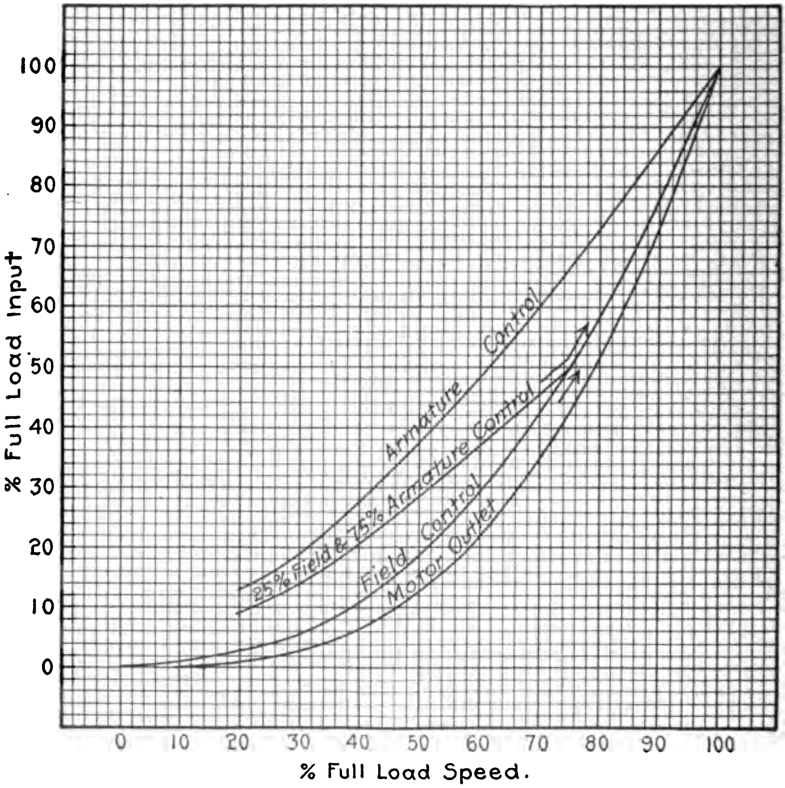


CHART NO. 4—CURVES SHOWING PER CENT. OF LOAD INPUT TO MOTOR DRIVING CENTRIFUGAL FAN AT VARIOUS PERCENTAGES OF FULL LOAD SPEED, WITH
ARMATURE CONTROL
FIELD "
25% FIELD—75% ARMATURE "

called "field control." The controller may also be designed for a combination of both armature and field control. A different type of motor is required for each of the three methods; that is, the same motor cannot be used in the three ways for a given fan. One of these methods must be selected in each particular case and should be determined from a consideration of the first cost as compared with operating costs.

From various tests which have been made, it has been found that the power required to drive a centrifugal fan varies approximately as the cube of the fan speeds. This is shown by the curve marked "Motor output" on chart No. 4. At 100 per cent. speed, or the rated speed, the output is considered as 100 per cent. As the speed is decreased to any per cent. of the rated speed shown by the figures at the bottom of the chart, the corresponding percentage of input or current consumption is given by the figures at the left of the chart. The other curves show the corresponding input or current consumption at various speeds for three different methods of speed control, namely, by armature control, field control and by a combination of 25 per cent. of the maximum speed by field control and the remainder by armature. The curve "motor output" shows the actual output or current consumption, assuming no losses in the controlling apparatus.

Chart No. 5. shows the percentage of control loss at various percentages of full load speed for the three methods of control, assuming that the loss in field control resistance is zero which is nearly true. From this chart, it can be seen that armature control is the most wasteful of current, as the current through the armature must be decreased by placing resistance in this line to decrease the speed, and this energy is wasted in the resistance.

A motor and control designed for field control is more expensive than one designed for armature control, but, as can be seen from the charts, it is more economical when operating at reduced speeds. If the approximate speeds and the periods of time through which the fan is to operate at the lower speeds are known, or can be assumed with any degree of accuracy, the most economical method of control can be calculated by figuring the interest on the increased cost of the one method against the reduced cost of current consumption. For fan work, it is seldom desirable to reduce the speed much below 25 per cent. of the maximum, and therefore the third method

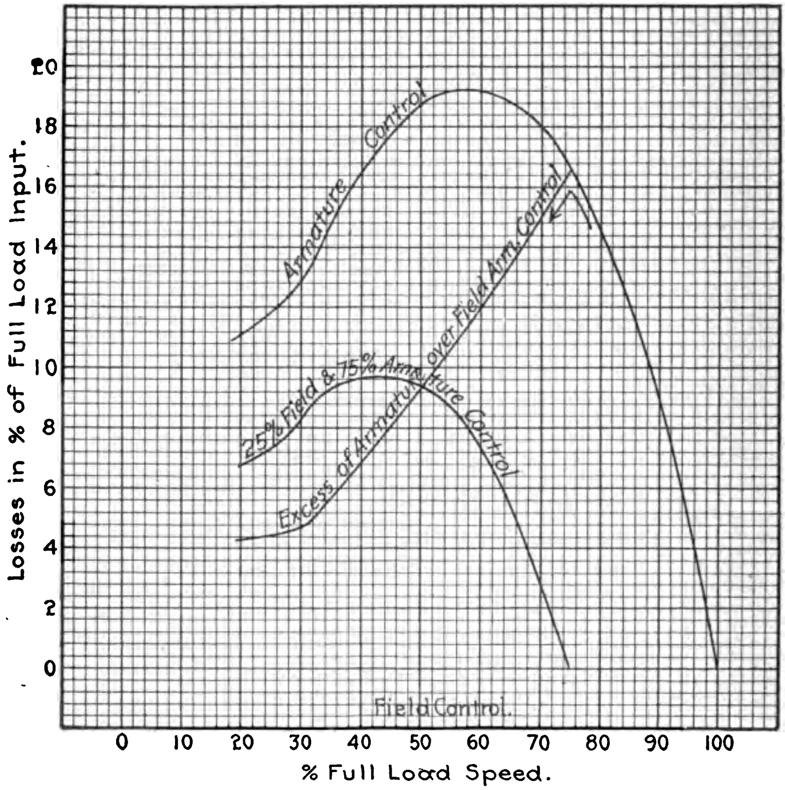


CHART No. 5—CURVES SHOWING CONTROL LOSSES IN PER CENT. OF FULL LOAD INPUT TO MOTOR DRIVING CENTRIFUGAL FAN AT VARIOUS PERCENTAGES OF FULL LOAD SPEED, WITH
ARMATURE CONTROL
FIELD "
25% FIELD—75% ARMATURE "

of 25 per cent. by field and 75 per cent. by armature is a good combination to adopt. The above curves apply only to motors in connection with fans, as the load variation would not be the same with other service.

Variable Speed Alternating Current Motors

The variable speed polyphase motor, commonly known as the phase wound or slip ring type, has proven very satisfactory for fan service, but generally speaking, the induction motor is not so well adapted for variable speed as the direct current motor, although its speed can be varied through a considerable range. This is accomplished by providing the rotor (armature) with a three phase Y connected winding and connecting a variable resistance in series with each phase. The three terminals of the winding are connected to collector or slip rings mounted on the shaft, and the rotor current flows by way of the rings through the resistance.

By means of a three pronged arm, the resistance in series with each phase of the rotor winding can be varied from the full amount to zero, thus varying the speed from minimum to full speed. The standard variation is 50 per cent. which is usually sufficient.

This method of speed control is very similar to armature control or direct current motors, and the resistance or rheostatic losses for this type of control are practically the same as the control losses of armature controlled direct current motors (See armature control curve—Chart No. 5).

It is impractical to build alternating motors to operate at the slow speeds characteristic of fan service; medium speed motors are used which operate the fans by either belt or silent chain drive, and if care is taken in selecting the proper motor, and in making the installation, the alternating current motor will prove nearly as satisfactory as the direct connected slow speed direct current equipment.

Fan and Motor Foundation

Where the system is required to operate noiselessly, or as nearly so as possible, the question of foundation is one of considerable importance. One of the best methods is to provide first the concrete or brick base in which are the foundation bolts. On this should be placed a yellow pine frame of proper dimensions, to accommodate the base of the fan and motor. This frame should be about 3 in. thick. On the top of this frame place a layer of compressed cork board 1 in. thick, covering the entire frame. The fan is then placed in position on this cork and secured by means of the bolts. An additional precaution should always be taken by providing a canvas sleeve about 6 in. wide between the fan outlet and main duct. This entirely separates the fan motor from the duct system so that no vibration can be communicated to the rooms.

CHAPTER XXIX

TEMPERATURE REGULATION

THE subject of temperature control and temperature regulating devices may properly be divided into two classes namely, direct acting and indirect acting. In both classes the same principle is involved and that is the expansion of substances due to the action of heat That part of the device which operates by expansion is called a thermostat. The class of direct acting would include all self-contained devices which contain within themselves the expanding or operating medium which acts directly upon the part to be controlled. Under the indirect class would be included all apparatus which employs the use of air, water or some fluid under pressure as the operating meduim, the expanding member simply opening to closing parts to allow the passage of this compressed fluid to or from the part to be controlled.

The direct acting thermostat is the simplest of the two kinds and is comparatively limited in its field of operation. The expanding medium adopted is invariably some form of liquid which has a low boiling point. The liquid is contained in a hermetically sealed chamber. As the temperature of the room in which the thermostat containing the liquid is placed, reaches the point for which the thermostat is set, the liquid begins to boil due to the heat of the surrounding air. This immediately raises the pressure within the chamber and it is this pressure that is utilized to accomplish the desired movement.

In some types the chamber containing the fluid is made in the form of a bellows which expands when the internal pressure is created. The movement of the bellows is increased by means of levers and transmitted to the part to be controlled by means of a chain or small wire cable and pulleys. In other types the pressure created is transmitted through a hollow tube to a diaphragm. The movement of the diaphragm is transmitted to the part to be controlled by means of a lever.

With the bellows type the bellows expands against the action of a spring which serves to force the bellows back to its original position when the temperature falls. This spring also serves as a means of adjustment of the temperature at which the thermostat acts. By increasing the tension of the spring, its pressure on the bellows is increased. This increased pressure necessitates a higher internal pressure for operation and a correspondingly higher temperature before the thermostat moves. The adjustment of the tension of this spring is usually accompanied by the movement of a small lever on an indicating dial which is calculated to read in temperatures, so that the thermostat can be set for any desired temperature within certain limits. The limit of operation of any one thermostat is about twenty degrees but they may be obtained to operate for this range at any point on the scale. The thermostat used to control the temperature of a living room operates between the limits of 60° and 80°. For dry room work, dry kilns, etc., they can be designed to operate as high as 160° and 180°. This is accomplished by simply changing composition of the fluid.

With the hollow tube type the regulation of the temperature is obtained by adjustable weights on the lever. Placing the weights at various positions on the lever exerts a heavier or lighter pressure on the diaphragm and necessitates a higher or lower temperature for operation.

The direct acting thermostats are most generally used on small residence installations and operate directly upon the check drafts of the boiler. The thermostat is located in the most central part of the building and if the bellows type is used, the chain or cable runs concealed and is attached to the lever controlling both the draft and check dampers. With the other type the bellows tube is run concealed to the diaphragm located directly over the boiler.

Fig. 61 shows a general arrangement of the apparatus. It can be seen from this that the boiler is controlled by the temperature of one room only. For ordinary residences, however, the rooms on the first floor are so open to each other that the variation in temperature in the different parts is very slight and above arrangement is satisfactory. It would not be well adapted, however, to large residences and buildings where the rooms are entirely separated and there is apt to be a variation in temperature with different weather conditions.

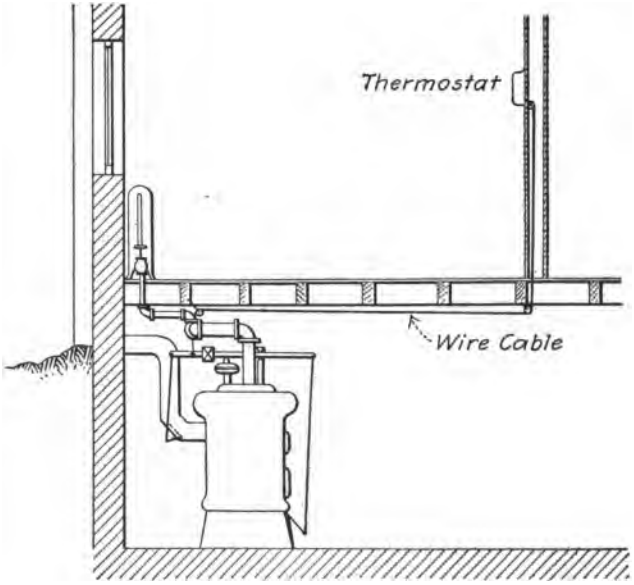


FIG. 61

Regulation for Hot Water Systems

The arrangement shown in Fig. 61 is not very satisfactory for gravity hot water systems as there is too much apparent lag between the fire and the room temperature due to the large volume of water to be heated. If the room temperature drops below 70° the thermostat operates and opens the drafts. The fire begins to burn rapidly and by the time the temperature of the water has been raised to such a point to heat the room sufficiently causing the drafts to close again the fire has burned to such an intensity that the water temperature and room temperature will continue to rise for a period after the dampers have closed.

Better results will be obtained by controlling the drafts directly by the temperature of the water. This arrangement, however, necessitates estimating the temperature of water that will be required with certain outside weather conditions and then setting the regulator for this temperature. With a little practice this can be done with very good results.

This same type of thermostat may be used in connection with ventilating systems where the air to all rooms is supplied for ventilating purposes only and at one temperature, usually 70°, the heat losses through windows and walls being taken care of by direct radiation in the rooms. The system was outlined under school and theatre ventilation. One method of accomplishing this temperature control is to install the thermostat in the main supply duct near the outlet of the fan and control the steam supply to the heaters.

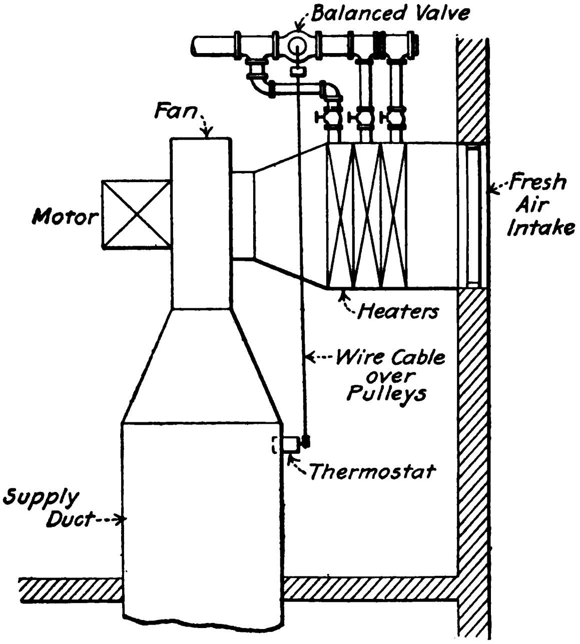


FIG. 62

The valve in the steam main operated by the thermostat must be the balanced double seated type and so adjusted as to work freely. The heater for this class of work is usually composed of three stacks. It is advisable to control only two of the stacks by the thermostat and equip the third with an ordinary gate valve. Each of the two thermostatically controlled stacks should have gate valves in their connections in addition to the balanced valve in the main connection. With this arrangement the one stack may be kept closed most of the time and used only in severe weather. In mild weather one of the remaining two may be shut off leaving only one stack in operation. Fig. 62 shows a plan view of this arrangement.

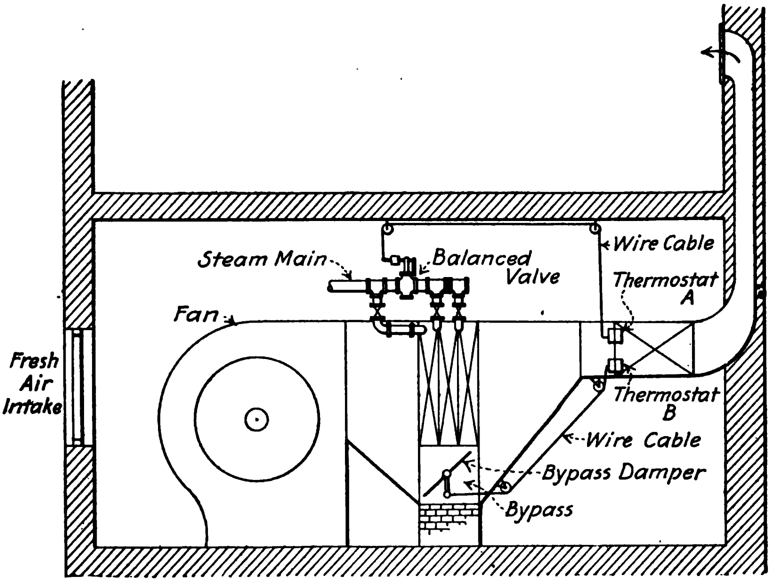


FIG. 63

The thermostat is so located within the duct that the air flowing through comes in contact with the expanding chamber.

Closer regulation may be obtained by the use of two thermostats as shown in Fig. 63 which is an elevation of a fan room. Here the thermostat A controls the steam supply in the manner as given in Fig. 62. Thermostat B operates the by-pass damper below the heater. Thermostat B should be set to operate at about one degree above A so that with a rising temperature the steam is first shut off from the heaters and in case the temperature continues to rise due to the steam remaining in the coils the by-pass damper immediately opens and allows sufficient cold air to enter, keeping the air at the required temperature.

This same system may be used in connection with the ventilation of auditoriums and theatres in the same manner. The direct radiation within the auditorium may also be controlled if this is desired. To do this the piping must be so arranged that all the radiation in the auditorium is supplied from a separate main and no other part of the building fed from this main. A balanced valve can be installed at some convenient point in this main and controlled by the thermostat located at a central point within the auditorium.

Indirect Systems

The chief difference in the various types of indirect control systems is in the design of the thermostat. The force to operate the valves, dampers, etc., is derived either from compressed air or water under pressure. Compressed air is most generally used as there is less danger of the valves and piping becoming clogged or corroded by its use than with water. A small air compressor and storage tank is necessary. The air compressor may be operated either by an electric motor or a hydraulic motor. A pressure of about 15 pounds per square inch is maintained in the tank.

The compressor is controlled automatically so that the pressure in the tank is at the proper amount. From this pressure tank a system of piping is run throughout the building to the various rooms to be controlled. The air line is run first to the thermostat which is located at some convenient point on an inside wall of the room. From the thermostat a line is extended to the diaphragm valve. If the room in question is heated by direct radiation, the diaphragm valve is placed directly on the steam supply to the radiator, the diaphragm being connected directly to the spindle of the valve which raises and lowers the valve seat. If there are two or more radiators in the room the air line from the thermostat may branch into as many lines as there are radiators and all radiators be controlled by the one thermostat. It is not advisable however to control more than five radiators by one thermostat or to allow one thermostat to control too large an area as there is apt to be considerable variation in temperature at different points and the results thus obtained might not be satisfactory.

If the room is heated by the indirect system with mixing dampers in the supply duct to the room, the air line from the thermostat is carried to a diaphragm motor which controls the mixing dampers by means of a lever. The thermostat is so constructed that when the temperature of the room is above that for which the thermostat is set certain ports are open which allow the air pressure in the system to be communicated through the air line to top of the diaphragm. This allows the full air pressure of the system, 15 pounds per square inch, to be exerted over the total area of the diaphragm. This pressure forces the diaphragm down which carries the valve spindle and seat with it, thus closing the valve and holding it in this position until the temperature in the room falls. With the lowering temperature the thermostat moves back in the opposite direction closing the port on the air supply main and opening a port in the line to the valve. This allows the air pressure on the top of the diaphragm to be released. A spring attachment below the diaphragm raises the valve spindle to its original position and opens the supply to the radiator. The same operation occurs when the thermostat controls the mixing dampers in the supply duct of an indirect or hot blast system. The diaphragm is attached to a lever which alternately opens and closes the dampers.

Types of Thermostats

There are two types of thermostats in general use. One type utilizes a fluid with a low boiling point in a sealed chamber as previously described as the expanding member. In the other type the expanding member is composed of two strips of metal with different coefficients of expansion, riveted or welded together. This member is fixed at one end and free to move at the other, as the temperature changes the two metals expand at different rates and this causes the strip to bend and thus opens or closes the ports. The thermostats may be still further divided into two separate types called the positive and the intermediate type. The positive thermostat is so constructed that when the temperature varies slightly from the neutral position one way or the other, the air ports are fully opened or closed and the diaphragm moves through its entire range which completely opens or closes the steam valve or the mixing dampers. The intermediate thermostat is so constructed that a slight variation of the temperature from neutral, only partially opens or closes the steam valve or moves the mixing dampers to an intermediate position and holds them in this position until there is a further variation in the temperature.

A section of the Johnson positive thermostat is shown in Fig. 64.

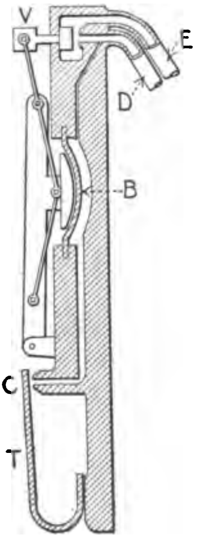


Fig. 64

The piece marked T is the expanding member. The metals used in constructing this strip are brass and steel. The line E is the air line to the radiator valve or the mixing damper. D is the air pressure line from the storage tank. With the expansion piece in the position shown the radiator valve would be open as full air pressure from D would pass through directly into the line and move the diaphragm down. There is a small leakage of air through the small auxiliary port to B when the thermostat is in this position but the amount is negligible. When the room cools slightly T moves back and closes the opening C. This allows the air pressure to accumulate back of the flexible diaphragm B through the small auxiliary part until this pressure is sufficient to move the diaphragm out carrying with it the knuckle movement. This movement causes the valve U to move in and close the air line D and allows the air pressure in E to escape around the valve spindle U to the atmosphere.

A section of a Johnson intermediate thermostat is shown in Fig. 65. When the port "C" is completely closed the full air pressure from "D" passes through the auxilliary port and collects on the diaphragm "B". This forces the main valve F down letting the compressed air from the supply line "D" pass through the chamber "H" into the chamber "G" as the valve is forced off its seat.. The air then passes from "G" through the line to the mixing damper. As the room temperature begins to rise due to the open position of the dampers, the thermostatic strip "T" moves a very slight amount off from the port "C." This allows the diaphragm "B" to move outward slightly and part of the air in the line "E" escapes. The diaphragms then take an intermediate position and are held thus until a further change in temperature occurs.

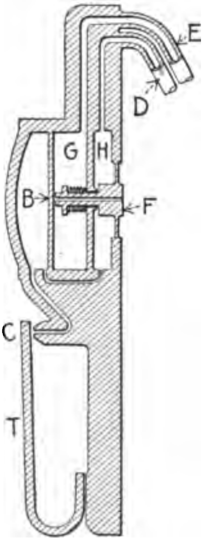


Fig. 65

The intermediate thermostat is used in connection with the ventilating systems of hotels, schools, etc., to control the mixing dampers when close regulation is desired.

Fig. 66 shows a cut of a set of mixing dampers with the diaphragm motor attached. These dampers are usually set so that the upper one controls the hot air and the lower one controls the cold or tempered air. The two dampers are connected by means of a link so that they both move together and they are always parallel to each other. The hot air and cold air ducts, however, unite at right angles so that when the damper in one is closed the other damper is open. With the dampers in the position as shown in the cut a portion of the air will flow in from each of the ducts with a resulting air temperature between the two. In some cases the cold

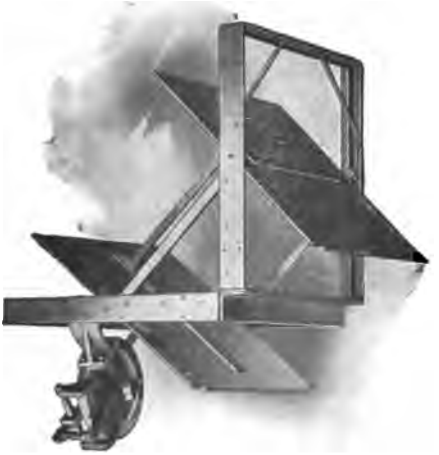


Fig. 66—Mixing Damper

air and hot air ducts run parallel and one above the other. In this case the dampers may be attached to one continuous spindle or trunion which passes up through the center of the two ducts. The dampers should then be set at right angles to each other on the spindle.

Fig. 67 shows a section of a diaphragm radiation valve that is commonly used. The valve body and seat are the same as any standard radiator valve. On the top of the valve above the rubber diaphragm is shown the connection for the air line from the thermostat. Below the diaphragm is the spring which raises the diaphragm after the air pressure has been released.

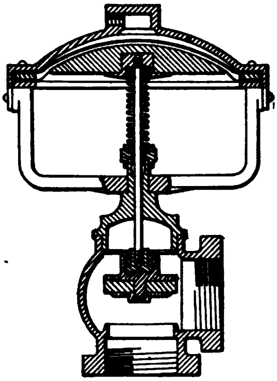
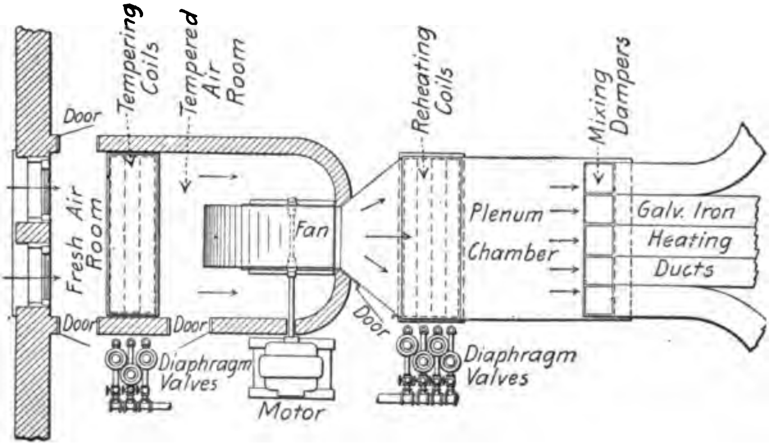


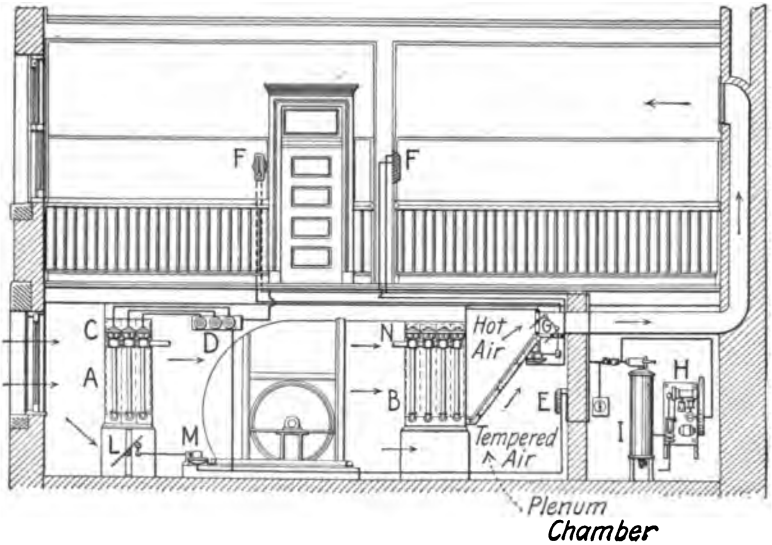
FIG. 67—CROSS SECTION ANGLE VALVE

A typical arrangement of a complete thermostatic control system in connection with an indirect heating and ventilating system is shown in Fig. 68. The tempering coil "A" is made up of three stacks and the reheater "B," four stacks. The cold-air enters at the fresh air opening and passing through the tempering coil is heated to a certain fixed temperature. This temperature is usually about 40° and is controlled by the thermostat "E" in the tempered air chamber. If the temperature of the tempered air goes above or below this point the thermostat operates on the coils and by-pass damper "L" through the relay valve "D" and thus keeps the temperature uniform. After passing the tempering coils the air passes through the fan and part of it then passes through the reheaters and the remainder through the by-pass below the reheaters, combining again after passing the mixing dampers "G" then on up into the various rooms.

This is called the individual duct system as each room supplied must have a separate duct from the reheater and a separate set of mixing dampers. The mixing dampers at "G" are controlled by the thermostat "F" in the room to which the duct leads.



Plan.



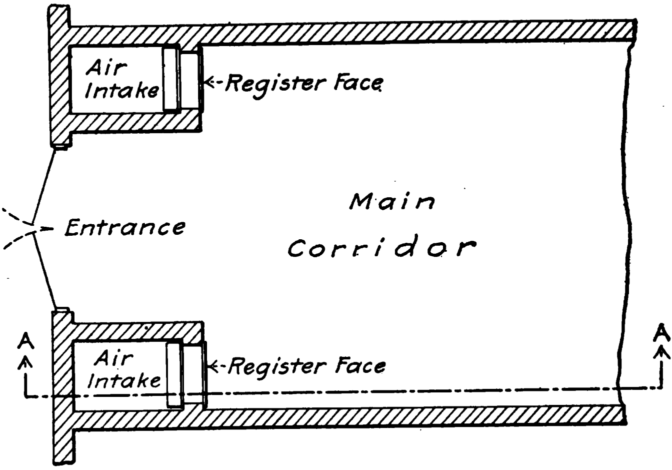
Elevation.

FIG. 68

CHAPTER XXX

RECIRCULATION OF AIR

IN all hot blast heating systems for factories and similar buildings a means for recirculating the air within the room should be provided. This is desirable in order to heat up the building quickly in the mornings. It will not only facilitate in heating the building quickly but also decrease the coal bill. Instead of drawing in the cold air from out of doors the air within the room is pulled through the heater to a considerable higher temperature than the outside air would be under the same conditions and thus the room is brought up to temperature with less time and less expenditure of heat units. It may also be possible to recirculate part of the air within the building during the daytime when the ventilating conditions or requirements are not severe and thus save in the coal consumption.

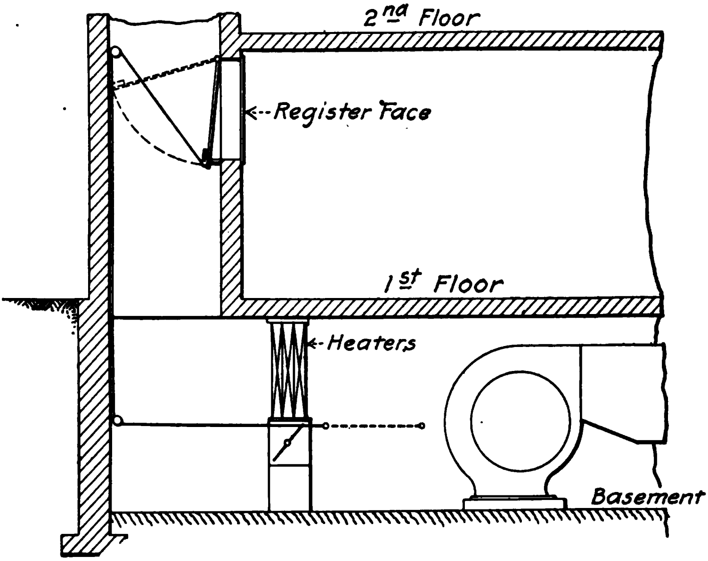


Plan of First Floor.

FIG. 69

To accomplish this recirculation a damper should be provided in the cold air inlet which can be closed completely. Between this damper and the tempering coils a door or opening the size of this cold air duct should be provided so that the air from the room will be drawn through the tempering coils. It may be necessary in some cases to extend from this opening, a recirculating duct when it is desired to draw the air for recirculation from some particular point.

Provision for the recirculation of air in school systems is also advisable from the standpoint of economy but there is always the possibility of the recirculating duct being left open at the time of school sessions, through carelessness on the part of the janitor or attendant.



Section A-A.

FIG. 69a

The air for recirculation in schools should be drawn from the main corridors if possible. This can usually be accomplished by means of doors and passages leading from the main corridors to the fan room. These doors will usually be closed during school hours and thus partly eliminate the possibility of recirculating the air at the wrong time.

Fig. 69 shows a method of recirculation when the air is drawn down from the roof through the fresh air ducts to the basement. Two fresh air ducts pass down the front of the building through the sides of the main corridor. A register face is installed in each of these ducts near the ceiling of the main corridor. At the back of the register face is a damper hinged at the top and weighted so as to normally remain closed. The damper is controlled by a chain from the fan room below and so arranged that it can be set in any position. If the damper is open to its full width the fresh air intake is completely closed so that all the air to the fan must be drawn from the corridor. If it is set in an intermediate position, part of the air is recirculated and the remainder is drawn from the outside.

CHAPTER XXXI

AIR WASHERS

THE subject of air washers and humidifying devices requires first a study of air under various conditions and the terms which apply to these conditions. The term "humidity" applies to the moisture or water vapor mixed with the air. There are two ways of referring to the humidity, namely, "absolute" and "relative." Absolute humidity is the actual amount or weight of water vapor contained in a cubic foot of air at a given temperature and percentage of saturation and is expressed in grains of moisture per cubic foot.

Relative humidity is the ratio of the weight of moisture contained in a given quantity of air to the total weight of moisture that this same quantity of air will retain when fully saturated. Air is said to be saturated when it contains the maximum amount of water vapor possible. The amount which it will hold in suspension depends upon the temperature. The higher the temperature the more water it will contain. Table No. 36 gives the maximum amount of water vapor in grains per cubic foot when fully saturated at various temperatures.

TABLE XXXVI

Degrees Fahrenheit	Grains per cu. ft.	Degrees Fahrenheit	Grains per cu. ft.
10	.78	70	7.98
15	.98	72	8.51
20	1.24	76	9.66
25	1.55	80	10.93
30	1.94	84	12.36
35	2.37	88	13.94
40	2.85	92	15.69
45	3.42	96	17.63
50	4.08	100	19.77
55	4.85	104	22.13
60	5.75	108	24.72
64	6.56	112	27.88
66	7.01	116	30.10
68	7.48	120	34.80

The absolute humidity may be expressed without reference to the temperature but in giving the relative humidity the temperature must be stated, as it is directly dependent upon the temperature. If the amount of moisture in a cubic foot of air is kept constant and the temperature varied the absolute humidity will be the same at all times, but the relative humidity will vary. To illustrate, assume a cubic foot of air at 40° F. From the above table it will be seen that this contains 2.85 grains of moisture. Its absolute humidity is expressed as 2.85 grains per cubic foot. Its relative humidity must be 100% as it is fully saturated. Let the temperature of this cubic foot of air be raised to 70° without allowing any moisture to be added or taken away from it and the absolute humidity will still be 2.85 grains per cubic foot. To find the relative humidity refer to the table and we find that at 70° one cubic foot of air will contain 7.89 grains when fully saturated. The relative humidity of the cubic foot of air containing 2.85 grains at 70 deg. will be $\frac{2\ 85}{7.98}=35.7\%$.

The dew point which is the basis of all calculations of humidity is the temperature at which air will be completely saturated with a given weight of moisture. If the temperature of this air is dropped below the dew point part of the contained moisture will be condensed and collect in the form of drops of water on the surrounding objects. Take a cubic foot of air under the same conditions as assumed above, containing 2.85 grains at 70°, under which conditions the relative humidity is 35.7%. If this air is cooled down again its relative humidity will increase until it reaches a temperature of 40° when it will have a relative humidity again of 100%. This will also be the dew point for air containing 2.85 grains per cubic foot. Assume now that it is cooled still further to a temperature of 30°. Its relative humidity must still be 100%, but from the above table it will be seen that at 30° and 100% humidity or complete saturation one cubic foot of air can contain only 1.94 grains of moisture per cubic foot. Therefore 2.85—1.94 = .91 grains of moisture will have to be condensed and form in particles of water. If this water is removed and the air again heated to 70 deg., its relative humidity will now be $\frac{1.94}{7.98}=24.4\%$ instead of

35.7% as before. It can therefore be seen that with proper apparatus the temperature and humidity of the air may be placed under absolute control.

Measurement of Humidity

There are several different methods and devices used for measuring humidity and nearly all of these depend upon the evaporation or rate of evaporation for their operation. When water or any other liquid is changed into a vapor, heat is required to accomplish this and the amount of heat required is the latent heat of the liquid at the temperature of evaporation times the weight of the liquid evaporated. The heat required for this evaporation is taken from the body upon which the liquid rests. This is plainly demonstrated by placing a small quantity of alcohol or ether upon the hand. A sensation of cold is immediately produced. This is caused by the rapid evaporation of the liquid which takes the heat from the hand to produce the evaporation. The same phenomenon occurs with water only the evaporation is less rapid and the sensation of cold is less noticeable. The dryer the air the more rapid will be the evaporation of water and the more rapid will the heat be taken from the adjacent object.

This is applied to the measurement of humidity by wrapping a piece of muslin tightly over the bulb of a thermometer and allowing one end of the muslin to rest in a receptacle containing water. This is called a wet Bulb Thermometer. The water is drawn up into the muslin due to capillary attraction and evaporates into the surrounding air. The evaporation of the water extracts heat from the bulb of the thermometer and causes it to read at a temperature below the actual temperature of the surrounding air. If the air is dry the evaporation will be rapid and the wet bulb temperature will be considerably below the dry bulb temperature.

If the air is thoroughly saturated or at 100% humidity there will be no evaporation from the muslin and the two thermometers will read alike. In order to have the instrument read accurately a rapid current of air should be passed over the wet bulb so as to remove the vapor as rapidly as it is evaporated. For this reason the Sling Psychrometer shown in Fig. 70 is the most accurate instrument to use. The wet and dry bulb thermometers are provided with a handle. The cloth is saturated with water and the thermometers whirled rapidly for a short period and the two thermometers read. Fig. 71 shows a Hygrophant which is another form of instrument quite commonly used, but to read accurately some form of fan should be used to agitate the air surrounding the wet bulb.

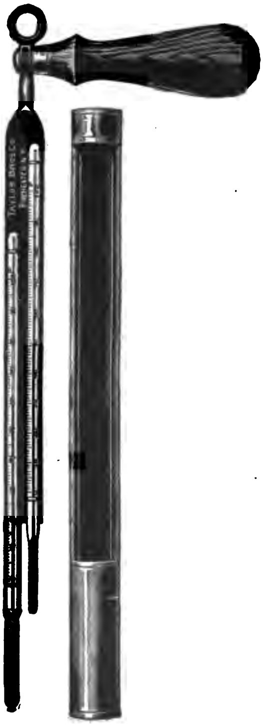


FIG. 70—SLING PSYCHROMETER

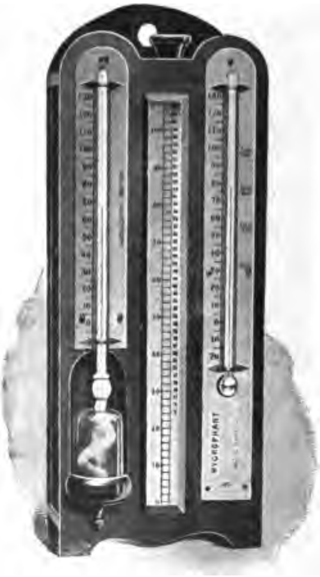


FIG. 71—HYGROPHANT

Fig. 70 shows another type called a Hygrodeik. This instrument is very convenient to use as the humidity may be read directly from the chart by means of the dial. With the others it is necessary to read the two thermometers and refer to charts to determine the humidity.

The difference between the two thermometer reading which is called the "Wet Bulb Depression" is the basis for determining the relative humidity. From the accompanying chart No. 6 given by the Carrier Air Conditioning Company the relative humidity may be determined with any wet bulb depression. At the bottom of the chart are given dry bulb temperatures from 20 degrees to 105 degrees F. The straight

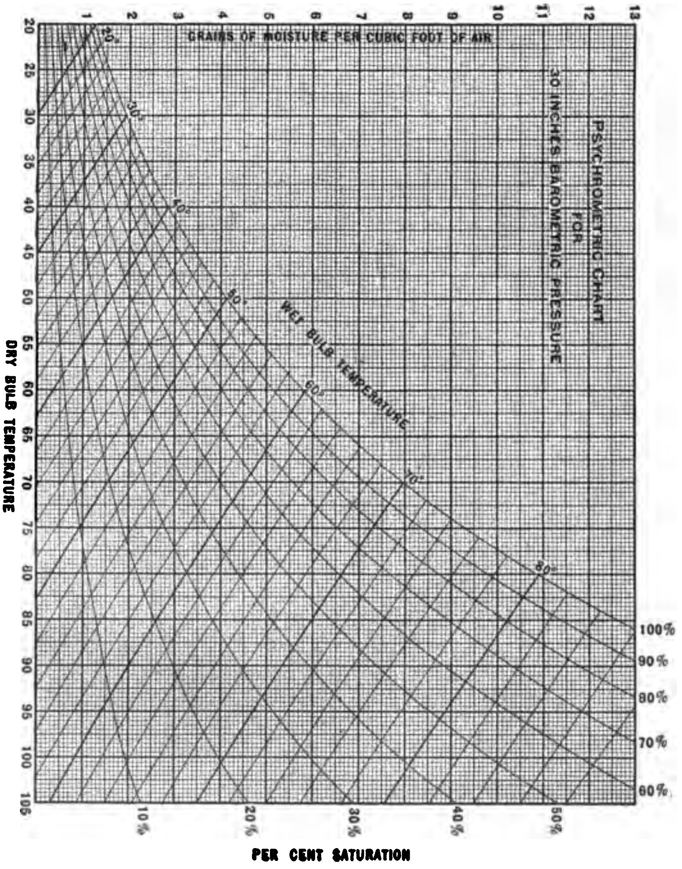


Chart No. 6

diagonal lines running up to the left are the wet bulb temperature readings and the readings are given at the top of these lines. The curved lines running up to the right are per cent. of saturation or relative humidity. At the left are given grains of moisture per cubic foot or absolute humidity. It will be noted that where the wet bulb temperature lines intersect the

100% humidity line these temperature readings are the same as the dry bulb temperatures which should be the case as at 100% humidity the two temperatures are the same.

Assume that with a Sling Psychrometer a reading of 70° is obtained on the dry bulb thermometer and 50° on the wet bulb temperature. What is the relative humidity and the grains of moisture per cubic foot?

From the 70 degree dry bulb temperature line follows up vertically until this line intersects the 50 degree wet bulb temperature line. This point of intersection is found to fall directly upon the 20% humidity line, therefore the relative humidity is 20%. From this same point of intersection follow horizontally to the left and the grains of moisture per cubic foot are found to be 1.6. If the dew point is desired under these conditions it can be determined by finding the point of intersection of this horizontal line with the 100% humidity line. From this point of intersection follow down to the bottom of the chart and the dew point is found to be at 26° F.

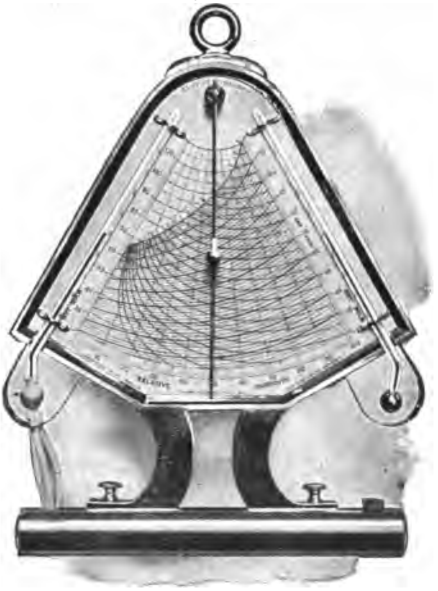


FIG. 72—HYGRODEIK

The effect of a low humidity is to cause the body to feel cold even in a comparatively warm room. This is because the dry air produces a rapid evaporation of the moisture from the body which, as has been demonstrated, requires heat and this heat is taken from the body producing a sensation of cold. A comparatively high humidity therefore not only produces more comfortable living conditions but also a healthier atmosphere. The best conditions are produced with a relative humidity of from 50% to 70%. With this humidity a room temperature of 65° to 68° will be found to be very comfortable. During the winter time the outdoor air will average from 40% to 70% humidity. Assume an outside temperature of 20° and a humidity of 50%, from the chart, the absolute humidity is found to be about 6 grains per cubic foot. If this air is heated to 70 degrees without the addition of moisture the relative humidity falls to $\frac{.6}{7.98} = 7.5\%$. Therefore the necessity for some means of humidifying the air can readily be seen. This is one of the prime objects of an air washer. The other is to remove the dirt and dust particles that are objectionable.

Nearly all the standard makes of air washers are constructed along similar lines, the differences being chiefly in the design of the spray nozzles. The apparatus consists of a rectangular sheet metal casing within which are arranged a series of spray nozzles which produce a sheet or fine spray of water over the entire cross-sectional area through which the air must pass. The water falls to a tank at the bottom, where it is drawn by means of a centrifugal pump, through a dirt strainer which removes the dirt and forced back again through the spray nozzles. The same water is therefore used over again. The amount which is carried away by the air is replaced by an automatic float valve in the tank, connected to a cold water line.

After the water passes water spray it then flows through a set of baffle plates which are called an eliminator. This eliminator removes the particles of water which are carried along by the velocity of the air, from the water spray.

The standard types of air washers require about eight feet in length for the entire apparatus and the cross-sectional area should be such that the velocity of air is about 500 feet per minute, not including the space required by the tank at the bottom. To illustrate, if the quantity of air handled is to be 50,000 cubic feet per minute the area of the washer should be $\frac{50000}{500}=50$ sq. ft. Assume a height of five feet, the width should therefore be $\frac{50}{5}=10$ feet. The floor area would be ten feet wide and about eight feet long.

For the height, allow about 18 inches for the height of the tank and five feet for the washer makes a total height of 6½ feet. The height and width can usually be adjusted to meet space conditions.

A tempering coil should always be placed in front of the air washer and should be arranged with about two or three stack deep. To estimate the size and depth of the reheating coils a drop in temperature of from 10 to 15 degrees should be assumed for the air passing through the washer due to the evaporation of moisture in the washer to raise the humidity to the proper amount. A space of at least two feet should be allowed between the washer and the tempering coil and also the reheater. Doors should be provided in the casings at these points for access to the interior.

Pumps

Centrifugal pumps are generally used for handling the water and are usually direct connected to motors. The head which the pumps discharge against in forcing the water through the spray nozzles is about 15 lbs. per square inch.

For the size and speed of pumps N. S. Thompson gives the following table No. 37 which applies to all standard types of machines.

TABLE XXXVII

Cubic feet per min.	Suction inches	Discharge inches	Speed R.P.M.	B.H.P.
5,000	2	1½	900	1¼
10,000	2½	2	900	1¾
15,000– 25,000	3	2½	750	2¼
30,000– 50,000	3	750	750	3½
55,000– 75,000	3¼	3¼	650	4
80,000–100,000	4	4	650	4½

Fig. 73 shows an air washer erected complete with pump connected. The action of the spray nozzles is also shown.

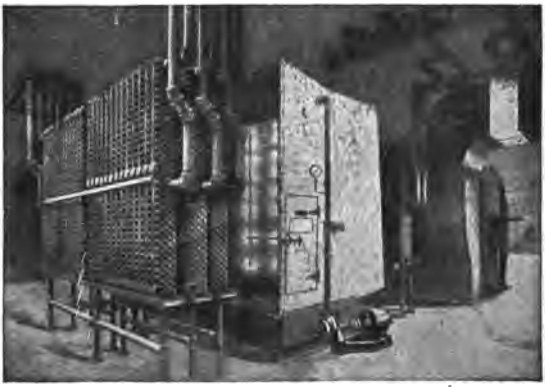


FIG. 73

At each end of the washer are shown the tempering coils and reheaters ready for the casings and sheet metal connections to the washer.

Fig. 74 shows a cut of a "Carrier" spray nozzle and Fig.

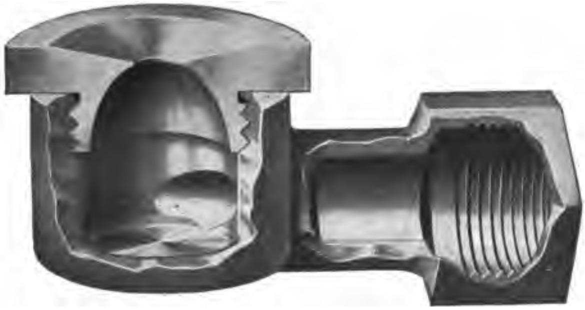


FIG. 74

75 shows the same in operation which gives some idea of the action of the spray.

Fig. 76 shows a cut of an "Acme" spray nozzle in operation and Fig. 77 shows the same with the spoon shifted to flush the nozzle. This operation is accomplished automatically at frequent intervals.

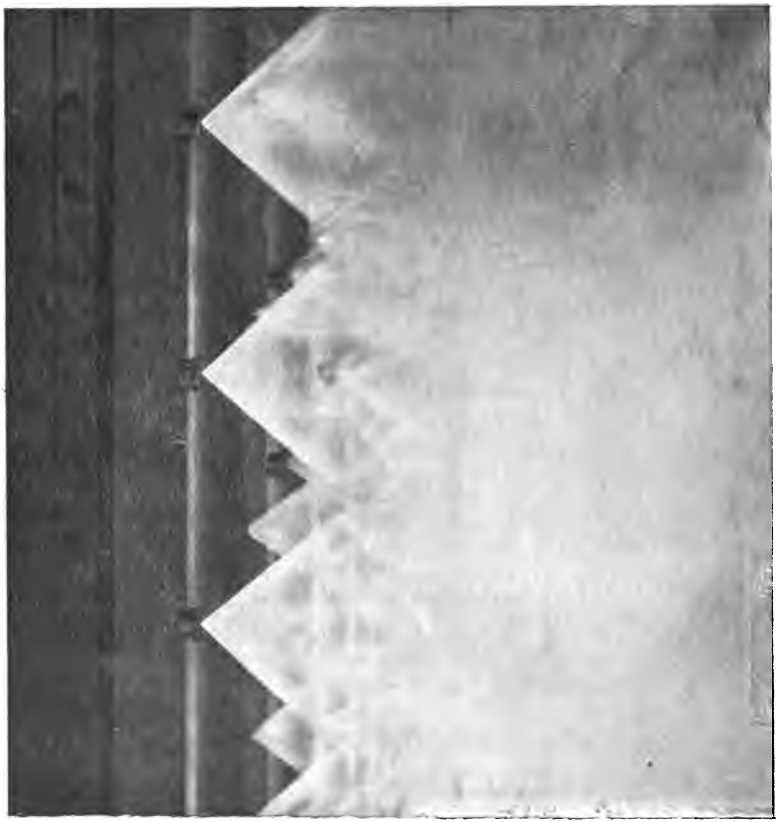


FIG. 75—SPRAY IN ACTION

Automatic Humidity Control

Automatic humidity control is accomplished by partially heating the incoming air and saturating it at this temperature by passing it through the water spray of the washer. The air is then heated up to the desired room temperature and in this process of heating above the temperature of saturation the humidity is lowered to the proper amount.

Assume that it is desired to maintain a temperature of 70°

and a humidity of 60%. Referring to the Chart No. 6 it is found that at the above conditions the air contains 4.8 grains per cubic foot. This horizontal line corresponding to 4.8 grains intersects the 100% humidity line at 54°, or the dew point for the above conditions is 54°. To maintain a condition of 60% humidity at 70° therefore, the air must at all times leave the air washer saturated and at a temperature of 54°. The air in passing through the reheater may be heated above 70° if desired to take care of the heat losses in the room. The

Fig. 76

Fig. 77

air would enter then the room a lower relative humidity than 60% but in cooling down to 70° the humidity would return to the desired amount of 60%.

The air leaving the washer is kept at the desired temperature by heating the water in the washer in connection with the tempering coils. The water is usually heated by means of a heater constructed of the principle of a steam ejector and may be supplied from the heating system if the pressure is above three pounds. Fig. 78 shows a type of heater used by the Carrier Air Conditioning Company.

The steam supply line to the heater is equipped with a diaphragm valve which is controlled by a thermostat located at the outlet of the air washer. This thermostat is set for the desired dew point temperature. Under the conditions as assumed above this would be 54°. The tempering coils should also be controlled thermostatically by two thermostats so that the air entering the washer will be kept constant at about the same temperature as the air leaving the washer. The water heater then supplies only the amount of heat that is required to evaporate the moisture to saturate the air.

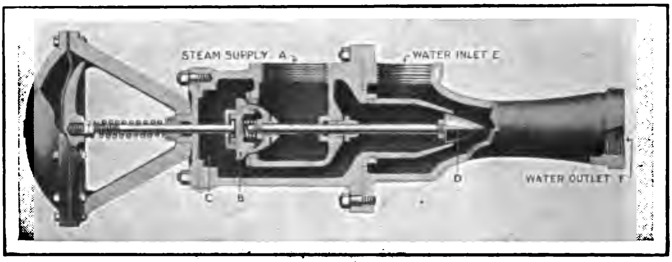


FIG. 78

A good arrangement for the thermostats and the tempering coils is to locate a cold air thermostat in the cold air inlet outside of the tempering coils. This thermostat should control the inside section or the one nearest to the air washer and set at about 20°. The second thermostat should be located between the tempering coils and the air washer controlling the remaining outside sections of the coil and set at about the same temperature as the water heater thermostat.

With this arrangement there is no danger of freezing the outer section of the coil. If the outside thermostat controlled the outside section as is often times done and set at 25° when the outdoor temperature goes above 25° the steam to this coil is shut off. At certain times water is apt to be held in the coil and will freeze causing more or less trouble. Under the arrangement suggested above this condition could not occur as steam will always be on the first coil when the outside temperature is below the freezing point.

CHAPTER XXXII

ESTIMATING COAL CONSUMPTION

IT is frequently necessary to estimate the coal consumption of a building for the whole or a certain portion of the time during which the plant will be in operation. One of the first questions asked of the designer is, "How much coal will this heating plant require?" This question is often asked even before the heating plant is designed or the size of the boiler plant known. There are several different methods of arriving at this desired information but all are more or less indefinite, as there are various things which can be only approximated. In cases where the amount of radiation or the necessary boiler capacity is not known the best method is to make an estimate on a basis of the cubical contents of the building to be heated. For buildings with average amount of exposed wall surface and normal conditions of heating, the coal consumption for heating alone will be from ¾ lb. to 1 lb. of coal for each cubic foot of contents heated per season, including Sundays and holidays. If the building is not heated during Sundays and holidays, the consumption will be from ½ lb. to ¾ lb. per season. This rule will be found reasonably accurate for average conditions and sufficiently accurate for all practical purposes. Allowances must be made for extraordinary conditions as to amounts of exposed wall surface, temperatures maintained, etc., and the rule modified to suit these conditions.

This rule does not allow for any mechanical ventilation and this must be estimated separately if part of the building has a mechanical ventilating system. This will of course increase the coal consumption proportional to the amount of air handled. Assume a hotel building 50 x 100 feet floor area and 12 stories high the height of the floors being 12 feet. The first two floors are to be equipped with a mechanical ventilating system which will supply about six air changes per hour for the entire space ventilated. How much coal would be necessary to heat and ventilate the building for the entire heating season?

To get the cubic contents of the building, the floor area is 50×100=5000 square feet. The height of the building is 12×12=144 feet or assumed 150 feet for round numbers. The total cube of the building is therefore 5000×150=750,000 cubic feet. The yearly coal consumption for heating alone without mechanical ventilation would be approximately 750,000 pounds or 375 tons. The space ventilated is 50×100×24= 120,000 cubic feet. With six air changes per hour, the air delivered would be about 120000×6=720000 cubic feet per hour or 120,000 cubic feet per minute. In the above calculation for coal consumption we have assumed this space heated by direct radiation which includes what ventilation would be obtained by natural means. This would normally be about one air change per hour and should be deducted from the air estimated for mechanical ventilation. The additional air to be delivered to the room is therefore 720000—120000=600000 cubic feet per hour, and it is only necessary to estimate this air heated to 70°. The average outside temperature for districts in the neighborhood of New York City and similar localities is about 45% to 50% of the extreme or assume the average temperature at 35°. The average range of temperature, therefore, through which the air for ventilation must be heated is 70°—35°=35°.

The average number of hours per day for operation of the ventilating system should be known and this can be approximated usually from the character of the building. Assume in the above case that the system will be operated ten hours per day and six days per week.

The heating season is usually from 220 days to 240 days per year including Sundays. In this case Sundays are not included in the operation of the ventilating system and the number of days for operation would be about 200. The total hours for operation would then be 200×10=2000 hrs. To find the average coal consumption per hour for ventilation the B. t. u. required per hour should first be determined. This will be the total air per hour times the range of temperature divided by 55 or $\frac{600{,}000\times 35}{55}=381{,}800$ B. t. u.

For average boiler efficiency each pound of coal will net 8000 B. t. u. per pound. Therefore $\frac{381,800}{8000}=47.8$ pounds of coal per hour. Total coal consumption for ventilation $\frac{47.8\times2000}{2000}=47.8$ tons per year. Assume 50 tons for round numbers. Total coal consumption for year per entire building is $375+50=425$ tons for heating and ventilation. This of course does not include coal necessary for hot water service, which should be estimated separately.

If the total amount of radiation to be supplied is shown, the coal consumption can be estimated on a B. t. u. basis in the same manner as outlined above for the ventilation.

To illustrate, assume the same building as above without any mechanical ventilation. The total direct radiation for this building would be about 9000 square feet estimated on a ratio of one to 80 or 85, square feet of radiation to cubic feet of contents. Estimating the radiation at 250 B. t. u. per square feet, the total B. t. u. required in zero weather would be $9000\times250=2,250,000$. For average conditions take 50% of this as stated above, which gives 1,125,000 B. t. u. per hour. Taking each pound of coal at 8000 B. t. u. as before the coal required will be $\frac{1,1250,000}{8,000}=140.6$ pounds per hour. Total heating season 220 days at 24 hours per day gives 5280 hours per season. Total coal per season is $140.6\times5280=742400$ pounds or 376 tons as against 375 tons estimated on a basis of cubic contents.

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